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Report 4070

EVALUATION OF A PNEUMATIC VALVING SYSTEM FOR APPLICATION TO A CIRCULATION CONTROL ROTOR

NAVAL SHIP RESEARCH AND DEVELOPMENT CENTER

Bethesda, Maryland 20034

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by

Kenneth R. Reader

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AVIATION AND SURFACE EFFECTS DEPARTMENT
RESEARCH AND DEVELOPMENT REPORT

May 1973

Report 4070

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Naval Ship Research and Development Center
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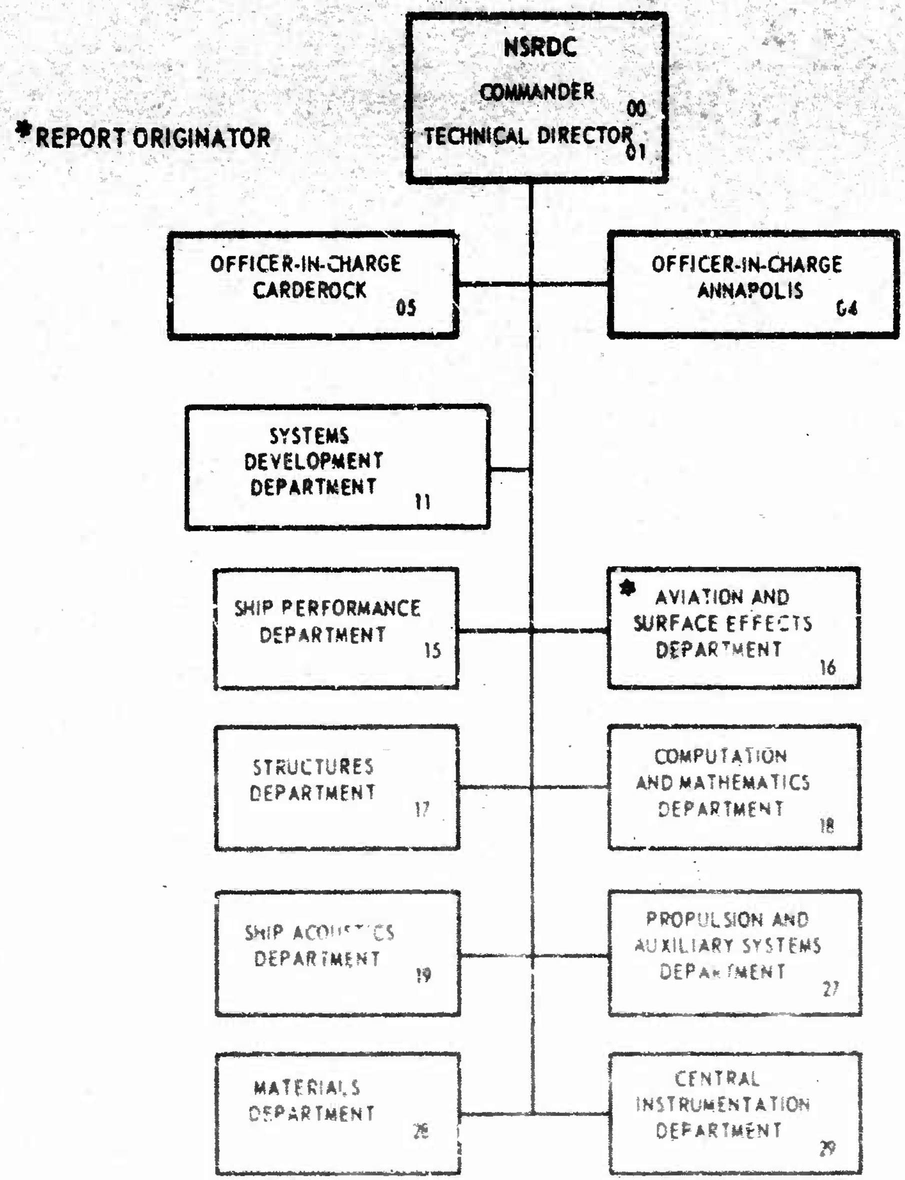

DEPARTMENT OF THE NAVY
NAVAL SHIP RESEARCH AND DEVELOPMENT CENTER
BETHESDA, MD. 20034

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NOTATION

DAP	Discrete azimuth position
h	Amount of nozzle covered by the cams, in inches
K_A	Nondimensional loss coefficient as defined in text
ℓ	Total length of pipe, in inches
one-per-rev	Occurring one time per revolution
p	Gage pressure, in pounds per square inch
$\bar{p}$	Average gage pressure, in pounds per square inch
two-per-rev	Occurring two times per revolution
x	Relative distance from entrance of pipe, in inches
x/ℓ	Nondimensional position along pipe
Δ	Peak-to-peak value (maximum minus minimum value)

Subscripts

end	End position (x/ℓ = 0.804)
ent	Entrance position (x/ℓ = 0.245)
mid	Middle position (x/ℓ = 0.524)
p	Pipe
$p\ell$	Plenum
s	Static
t	Total or stagnation

ABSTRACT

A cam-type pneumatic valving system has been developed to provide helicopter rotor control/trim forces. This valving system provides both first and second harmonic rotor control by means of modulating both blade pressures and mass flow rates. Data are presented for (1) constant, one-per-rev, and two-per-rev air modulation, (2) constant and tapered slot distributions, (3) two pipe volumes, and (4) three cam-nozzle gap distances.

The present study demonstrated that air pressure and mass flow rate can be modulated by means of a simple cam valve system. As the gap between the periphery of the cams and the nozzle was increased for a given cam geometry, the mean pressure and mass flow rate increased and the peak-to-peak pressure and mass flow rate decreased. It was also demonstrated that a smooth transfer of the total pressure and mass flow rate occurred in going from a one- to a two-per-rev component (or vice versa).

ADMINISTRATIVE INFORMATION

The work reported herein was sponsored by the Naval Air Systems Command (NAVAIR 320). Funding was provided under Project F41.421.210, Work Unit 1–1690–100.

INTRODUCTION

The objective was to evaluate the feasibility of modulating and controlling the flow of air to a ventilated helicopter blade by means of a simple cam valving system.[1] The actual air valve described herein is a working mockup of one installed in a 7-ft-diameter model rotor. When applied to a helicopter rotor, this valving mechanism will allow for control of quantities of first harmonic, second harmonic, and steady-state air flow. It will also provide a means of controlling the phase angle between the first and the second harmonic air flow and some fixed reference system to allow for azimuthally locating the maximum rotor air flow.

A system of cams and nozzles are provided to accomplish these regulating functions. The nozzles are designed into the rotor hub to facilitate passage of air from the hub plenum into each rotor blade. The cams are located in the hub opposite the nozzles; they are fixed with respect to the airframe, but their orientation is adjustable. Thus, the air valve is considered as a system of nozzles that rotate around stationary cams.

The valving mockup was evaluated in various geometric configurations to determine gap height effects, mean blade pressures, and associated loss coefficients. The valving system was run with varying degrees of

[1]Norman, G. and K. R. Reader, "Cam Type Air Control Valve," Patent application, Navy Case 53,968 (3 Aug 1971).

one-per-rev and two-per-rev inputs. Two receiver volumes (simulating internal rotor blade volume) were evaluated, and the influence of variation in blade slot height was determined.

BACKGROUND

As part of an ongoing effort to provide a better, more efficient helicopter, the Aviation and Surface Effects Department of the Naval Ship Research and Development Center (NSRDC) has been investigating a tangentially blown airfoil design which can readily be incorporated into a helicopter rotor blade. This shaft-driven rotor system developes control forces by modulating air mass flow and pressure to each rotor blade. Therefore, a simple, reliable valving system was required to provide the necessary air regulation. Two existing valving systems were considered:

1. A British system, developed by Cheeseman,[2] wherein the cyclic and collective control valves are positioned in the nonrotating part of the rotor hub. The air mass is fed through a shaft to an annulus divided into 24 segmental passages each of which contains a slide valve. Each blade samples part of the segmental passages. The valve opening is controlled by push rods. These are connected to the periphery of a control disk (swashplate) which can be raised, lowered, or inclined about any axis. This system is unable to include higher harmonic control unless a sophisticated electromechanical servo system is added to drive the swashplate mechanism.

2. The Lockheed system[3] wherein two separate air supplies are used to control the air on the advancing and retreating sectors of the rotor disk. In addition, each blade has two separate plenum chambers which supply air to the leading edge and trailing edge slots, respectively. When a blade is in the advancing half of the rotor disk, air from the advancing supply is directed through its trailing edge plenum to the trailing edge jet slot. (A manifold blocks the flow of air to the leading edge jet.) When a blade is in the retreating sector of the disk, air is ducted from the retreating air supply through both leading and trailing edge blade plenums to their respective slots. Roll trim is accomplished by adjusting the advancing and retreating air flows. The advancing blade pressure is analogous to conventional mechanical collective pitch, and the differential pressure between advancing and retreating sectors behaves like mechanical cyclic pitch. Inclusion of higher harmonic control capability for this mechanical system would be nearly impossible unless some differential higher order air modulation system could be created.

These extremely complicated valving systems have provided the stimulus to develop a more practical approach. The technology described herein meets the basic requirements of simplicity and higher harmonic control.

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[2]Cheeseman, I. C., "Circulation Control and Its Application to Stopped Rotor Aircraft," Lecture to Tenth Anglo-American Aeronautical Conference, Los Angeles (18–20 Oct 1967).

[3]Kuczynski, W. A. and R. B. Lewis, II, "Jet Flap Cyclic Twist Feasibility Research Program," Prepared for the Naval Air Systems Command under Contract N00019-68-C-0285 (31 Mar 1970).

MODEL AND EQUIPMENT

MODEL DESIGN

The air valve described herein is a mockup of a system intended to regulate the supply of air to a set of ventilated helicopter rotor blades through various air nozzles and cams. The air nozzles are mounted inside the rotor hub (one nozzle per blade) and rotate with it. The nozzle entrance openings are all equidistant from the center of rotation of the hub. Two cams, designated as first and second harmonic cams, are located in the hub in opposition to the nozzles. They remain stationary with respect to the rotating hub but are adjustable to provide maneuvering forces.

The first harmonic cam is proportioned so that the distance between its periphery and the nozzle face is a sine function of the angular position of hub, and the second harmonic cam is proportioned so that the distance between its periphery and the nozzle face is a sine function of twice the angular position of the hub. Thus, the first harmonic cam has a single lobe and the second harmonic cam has two lobes.

Since the two cams are jointly in opposition to the nozzles, the amount of air flow controlled by each cam is a function of the amount of nozzle opening that each covers. The axial adjustment of their positions provides selection of the relative amounts of air controlled by each. The quantity of steady flow air is regulated by adjusting the amount of separation (in the axial direction) between the two cams. This separation is accomplished by axial adjustment of either or both cams. The azimuth of maximum air flow is accomplished by the angular adjustment of both cams, as a unit or individually, through a 0- to 360-deg adjustment range. As a consequence, the phase angle between the first and second harmonic cams is also adjustable.

Changes in air mass flow rate are accomplished by regulating the pressure of the air supply to the hub through a controllable air pressure regulating valve inserted in the air supply line.

MODEL CONSTRUCTION

The mockup of the rotor valving system was constructed of wood, metal, and plexiglass. The main components consisted of a plenum, a one-per-rev cam, a two-per-rev cam, a single nozzle leading into a plugged pipe with an adjustable slot, and a variable-speed drive motor. Since this was to be an inexpensive model to demonstrate that the air flow can be proportionally modulated in a one- and a two-per-rev manner, it was decided to simplify the model by using rotating cams and a stationary plenum. Compromises were also made both in the material and in the accuracy within which the model was constructed. The plenum was constructed from plywood with a removable plexiglass top. One side was constructed so that a movable nozzle could be mounted to the plenum.

The wooden nozzle fitted into a plugged brass pipe which had an adjustable slot. The slot sides were parallel and the slot height was adjusted to 0.042 in. by means of external clamps. The pipe was designed to have a resonance frequency of 136 Hz and had three total pressure and two static pressure taps. The cams were constructed by gluing paper templets to pieces of hard wood and then sanding to shape on a large disk sander. The cams could translate along the driveshaft, making them positionable with respect to the nozzle;

provisions were made for locking them to the shaft. The shaft was belt driven by a 0.5-hp direct-current motor with adequate controls to regulate the rpm accurately.

The internal volume of the pipe will affect the buildup and decay of the pipe pressure, and thereby it can affect the losses through the cam-nozzle valve system. The total and static pressure distribution within the pipe will also be affected by the internal pipe volume. To estimate the influence of the internal pipe volume on the losses and the pressure distribution within the pipe, a filler rod was manufactured and used to arbitrarily reduce the pipe volume by 33 percent. The rod cross section was crescent shaped and equal to the pipe length. It was designed to have minimal effect on the pressure taps in the pipe.

The nominal gap clearance between the cam and the nozzle can serve to introduce a collective amount of air flow into the pipe. Two arbitrary gap clearances were selected to determine the attenuation of the peak-to-peak pipe pressure as the nominal gap was increased. To enable adjustment of the gap between the periphery of the cams and the nozzle, two wooden spacers, 0.25- and 0.50-in. thick, were successively installed between the side of the plenum which held the nozzle and the four adjoining sides. In addition, these spacers increased the volume of the plenum.

INSTRUMENTATION

The plenum pressure for all but a few runs was measured on a differential pressure gage (Wallace and Tiernan) calibrated in inches of mercury. The pipe total pressures and static pressures were measured by means of ±15-psid Statham pressure transducers. The pipe total pressures were measured at pipe x/ℓ's of 0.245, 0.524, and 0.804; the static pressures, were measured at x/ℓ's of 0.245 and 0.804. The stagnation temperature was measured in the plenum and at x/ℓ's of 0.245 and 0.804. The rpm was measured by a 60-tooth steel gear running in close proximity to a magnetic pickup.

The signals from the pressure transducers were amplified and then filtered by a 147-Hz filter before being recorded on an 18-channel oscillograph. The rpm was also recorded on the oscillograph. The amplifiers were calibrated for the phase shift and the amplitude attentuation caused by the filters. The only data that needed to be corrected for amplitude attentuation were those for the higher frequency two-per-rev configurations.

The general arrangement and principal dimensions of the model are presented in Figure 1. The experiimental arrangement is shown in Figure 2.

PROCEDURE

In the period from 28 January to 19 February 1969, the control valving system for a circulation controlled rotor was investigated to prove the feasibility of modulating the internal blade pressure in a one- and two-per-rev manner. The test consisted of systematically changing the percentage of the one-per-rev component, the two-per-rev component, and the constant flow of air for various configurations and then varying the plenum pressure and rpm. The configurations tested with various combinations of the one- and two-per-rev cam included 100- and 67-percent pipe volumes, 0.25- and 0.50-in. spacers between the nozzle and the

cams, and a linearily tapered slot. The tapered slot went from zero at the inboard blade station ($x/\ell = 0.32$) to a maximum height of 0.084 in. at the tip.

Pipe pressure measurements were obtained for a configuration by setting a plenum pressure and running a range of rpm's. This procedure was performed for plenum pressures ranging from 1 to 13.75 psi. The range of rpm's tested was from 0 to 3600, corresponding to frequencies from 0 to 60 Hz for the one-per-rev cam and from 0 to 120 Hz for the two-per-rev cam. The test conditions are listed in Table 1.

A discrete azimuth position method was an important technique used in testing the control valve. The cams were positioned and restrained while data were recorded. Then the cams were repositioned in increments of 30 deg, each time recording data for the nonrotating condition. A quasi-dynamic representation of the valve system was obtained in this manner. An evaluation of the method is presented later in this report.

RESULTS AND DISCUSSION

EFFECTS OF ROTATING FREQUENCY ON PEAK-TO-PEAK TOTAL PRESSURE

Figure 3 indicates the effects of the rotating frequency on the peak-to-peak total pressure for plenum pressure ranging from 1 to 13.75 psi. The one-per-rev cam configuration, Figure 3a, showed a reduction in the peak-to-peak total pressure as the frequency was increased. This trend was much greater for the higher plenum pressures of 13.75 psi. Another important observation is that for plenum pressures below 3.04 psi, the pressure wave for a given frequency was attentuated as it propagated down the pipe; for plenum pressures of about 8.84 psi, the pressure wave was attentuated or amplified depending on the frequency.

The two-per-rev cam configuration, Figure 3b, showed a general tendency for the total pressure peak-to-peak value to increase slightly as the frequency was increased. However, for lower plenum pressure, it exhibited the same trend as the one-per-rev cam configuration in that for a plenum pressure of 2.98 psi, the pressure wave was attentuated only whereas above that level, the pressure wave was attentuated or amplified depending on frequency.

When the peak-to-peak pressures are plotted versus the plenum pressure for constant frequencies, the crossing of the entrance curve with the middle and end station curves indicates the plenum pressure above which the pressure wave was attentuated and below which the pressure wave was amplified. These values are given in Table 2 together with a list of the relevant frequencies. These data are for both the one- and two-per-rev cam configurations and show the attentuation and amplification for both the center and end positions relative to the entrance position.

AVERAGE MASS FLOW RATE FOR VARIOUS CONFIGURATIONS

The average mass flow rate of the mockup control valve is presented in Figure 4 as a function of the plenum pressure for all configurations tested.

The effect of varying the percentage of one- and two-per-rev cam-nozzle exposure can be demonstrated by cross plotting the average mass flow rate and the percentage of the one-per-rev (two-per-rev) cam

TABLE 1 – TEST PROGRAM FOR A PNEUMATIC VALVING SYSTEM

Run Number	Speed Range rpm	Plenum Pressure Range psig	Slot Height in.	Cams: One-per-Rev percent	Cams: Two-per-Rev percent	Cams: Constant percent	Configuration: Volume percent	Configuration: Spacer in.	Configuration: Slot
1– 7	0–3600	2.94–13.75	0.042	0	100	0	100	0	Straight
8–17	0–3600	1.50– 8.00	0.042	50	50	0	100	0	Straight
18–22	0–3500	1.00– 9.00	0.042	33	33	33	100	0	Straight
23–26	0–3500	1.00– 9.00	0.042	67	33	0	100	0	Straight
27–29	0–3000	1.00– 9.00	0.042	75	25	0	100	0	Straight
30–32	0–3000	1.00– 9.00	0.042	33	67	0	100	0	Straight
33–37	0–3000	1.00– 9.00	0.042	20.8	79.2	0	100	0	Straight
38–41	0–3000	1.00– 9.00	0.042	100	0	0	100	0	Straight
42–46	0–3600	1–13.75	0.042	100	0	0	100	0	Straight
47–50	0–3000	1–13.75	0.042	100	0	0	67	0	Straight
51–56	DAP*	8.84	0.042	100	0	0	67	0	Straight
57–58	0–3000	2– 6.5	0.042	100	0	0	100	0.50	Straight
59–62	0–3000	1– 6.88	0.042	100	0	0	100	0.25	Straight
63	DAP	6.88	0.042	100	0	0	100	0.25	Straight
64–67	0–3000	1– 6.88	0.042	0	100	0	100	0.25	Straight
68	DAP	6.88	0.042	0	100	0	100	0.25	Straight
69	0–3500	6.88	0.042	0	100	0	100	0	Straight
70	DAP	6.88	0.042	0	100	0	100	0	Straight
73–80	0–3000	1–13.75	0–0.084	0	100	0	100	0	Tapered
81	DAP	8.84	0–0.084	0	100	0	100	0	Tapered

*DAP is discrete azimuth position.

TABLE 2 – EFFECT OF PLENUM PRESSURE ON THE PRESSURE WAVE ATTENTUATION AND AMPLIFICATION FOR VARIOUS FREQUENCIES

Configuration	Frequency Hz	Plenum Pressure*	
		Middle Station	End Station
2/Rev	34	N/A**	N/A
	50	N/A	N/A
	69	N/A	11
	83	N/A	6
	102	13.5	7
	120	12.6	7
1/Rev	18	N/A	N/A
	34	N/A	14
	51	11.5	9

*Below these values, the pressure wave is attentuated and above them, the pressure wave is amplified.

**N/A = no amplification, only attentuation.

exposure over the nozzle inlet for particular plenum pressures. This shows that as the cams were transferred from all two-per-rev to all one-per-rev or vice versa, the relationship between the average mass flow rate and the percentage of one-per-rev cam exposure increased in a nearly linear fashion for a constant plenum pressure.

An estimation of the effect of the slot area distribution on the average mass flow rate can be obtained from curves in Figure 4 by comparing the 100-percent two-per-rev cam-nozzle exposure for the uniform straight slot height and the uniform tapered slot height. The slot area was redistributed by reducing the inboard slot height to zero, doubling the outboard slot height, and linearly tapering the slot height in between. After the slot height was redistributed, there was a 2-percent reduction in total slot area. A comparison of these curves (Figure 4) indicated that the uniformly tapered slot configuration attained a reduction in the average mass flow of approximately 15 percent.

The effect on the average mass flow rate of varying the internal volume of the pipe can be noted from the curves of Figure 4 by comparing the one-per-rev cam-nozzle exposure for the 100- and the 67-percent volumes. Note that the addition of the wooden insert to reduce the internal pipe volume by 33-percent caused an average mass flow decrement of 0.0002 slug/sec which, depending on the initial average mass flow, represented a 2- to 5-percent reduction in the average mass flow rate from the initial volume.

To determine the effect of an increase in the gap between the cams and the nozzle, gap widths of 0.01 (nominal clearance), 0.26, and 0.51 inches were used. As shown in Figure 4, when a gap width of 0.26 in. was used, the 100-percent curves for the one- and two-per-rev cams were much closer. This was accompanied by a large increase in the average mass flow rate.

Increasing the gap to 0.51 in. increased the average mass flow rate by only a small amount. Since the pressure modulation was not present for this configuration, the gap width of the 0.51-in. curve therefore represents the maximum average mass flow rate that can be obtained for a given plenum pressure with this nozzle and slot area combination.

The control area of the cam-nozzle combination is defined as an area equal to the periphery of the nozzle times the gap between the cam and nozzle. A general overview of Figure 4 indicates that as the control area of the nozzle was increased, the average mass flow increased in a smooth predictable way that was characterized by an increasing slope.

A comparison of the actual and the theoretical area variation with azimuth position is presented in Figure 5 for the one- and the two-per-rev cams. The one-per-rev cam (Figure 5a) was constructed fairly accurately, but the two-per-rev cam (Figure 5b) was not. These inaccuracies did not hamper the main objective of the test, namely to demonstrate the feasibility of the cams for modulating the total pressure.

The mass flow rate is presented in Figure 6 as a function of the total nozzle control area for both the one- and the two-per-rev cams. The nozzle control area above the value of 1 in.^{2} was obtained by increasing the nominal gap of 0.01 to 0.26 in. As the nozzle control area increased, the mass flow rate for a constant plenum pressure became constant and approached a value which was maximum for the open nozzle slot area combination. For example, in Figure 6a the mass flow rate became asymptotic to a value of 0.011 slug/sec as the nozzle control area increased for a hub pressure of 6.88 psi. The significance is that for a

given plenum pressure, the mass flow rate is a uniform predictable function of the total nozzle control area and does not seem to depend on how the control area is varied, i.e., by varying the one- or the two-per-rev, by varying the width of the gap between the cams and the nozzle, or by a combination of the two. Compare the data of the one- and two-per-rev configurations for a plenum pressure of 6.88 psi shown in Figures 6a and 6b.

Figure 6a also indicates the effect of reducing the pipe volume by 33 percent for a plenum pressure of 8.84 psi. A reduction in mass flow rate was noted for the reduced volume configuration at a control area of 0.3 in.^{2}; it became larger as the control area increased. This suggests that no corrections need be made for a volume change in the receiving pipe for nozzle control areas of less than 0.3 in.^{2}. Above this control area, the data require correction, but information is insufficient to determine the correction qualitatively.

The dashed extension line for a plenum pressure of 8.84 psi (Figure 6a) was obtained by using the shape of the curve for a plenum pressure of 6.88 psi and shifting the curve by an amount determined from Figure 4 for the same configuration.

When the slot area of the pipe was redistributed, the curve coincided with that presented in Figure 6b although the plenum pressure was 8.84 instead of 6.88 psi. The point symbols were omitted from the graph for simplicity.

TOTAL PRESSURE LOSS DUE TO AREA CHANGE

The mechanical principle used to modulate the total pressure in the pipe was to introduce variable losses between the plenum and the pipe in a controlled and predictable fashion. These losses can be created by the use of a cam system such as described in this report.

Any change in flow direction, cross section area, or cross section shape will cause a loss in total pressure. In general, when dealing with these types of losses, the governing equation is:

$$\Delta P_T = K_A \frac{1}{2} \rho V^2 = K_A q$$

where K_A is the loss coefficient and q is the reference dynamic pressure. The dynamic pressure is generally evaluated at the reference area of the loss coefficient. For this application however, it was more convenient and useful to use the pipe entrance as a reference point. Thus ΔP_T is the plenum total pressure minus the pipe inlet total pressure and the dynamic pressure is at the pipe entrance. Therefore

$$K_A = \frac{P_{t_{p\ell}} - P_{t_p}}{P_{t_p} - P_{s_p}}$$

It is recognized that there is a possibility that the loss factor might be dependent on pipe configuration because of the way the values are defined in the definition of the loss factor K_A. Although slight, this dependence will show up later in the data; however, the data collected were insufficient to evaluate this effect either quantitatively or qualitatively.

It should be emphasized that the loss coefficient data presented in this report are for the nozzle completely covered by either the one- or the two-per-rev cam. Therefore, when combinations of other cam positions are used, the loss coefficient of the system will not necessarily be the sum of the individual cam loss coefficient; it may be either larger or smaller.

Figure 7 shows the inverse of the loss coefficient K_A^{-1} as a function of the total nozzle control area caused by the one- and two-per-rev cams. The considerable scatter in the data is attributed to manufacturing tolerances.

Data for the condition where the one-per-rev cam covered the entire nozzle control area (Figure 7a) were taken for the cam in proximity to the nozzle with nominal clearances of 0.01, 0.26, and 0.51 in. at the minimum gap and with the 100- and the 67-percent pipe volumes. The data for the configurations with 0.26- and 0.01-in. gaps can be combined to construct a fairly uniform curve for the 100-percent volume. By combining data of two different plenum pressures into a single curve, we imply that Reynolds number effects are minimal. Although this is not proven by the data presented here, additional work performed by the author indicates that there are no Reynolds number effects below a plenum pressure of 14.0 psi.

Except where noted in Figure 7a, the 100-percent volume of the pipe was used for Figures 7a and 7b. If Reynolds number effect is ignored, the data represented by the one- and the two-per-rev cams can be combined to give one curve that relates the inverse of the loss coefficient to the nozzle control area. Indications are that the loss coefficient is a predictable function of the nozzle control area for a given configuration. When the pipe volume was reduced by 33 percent, the loss coefficient was reduced considerably, as shown in Figure 7a. The total pressure measurement for the full volume configuration was taken at the center of the pipe, and it was assumed that the internal streamlines were similar for each configuration. When the pipe volume was reduced by inserting the wooden filler block, the resulting reorienting of the fluid streamlines may have caused a relative error in the measurement of the total pressure. No pressure survey was made to check the pressure profile with the insert installed in the pipe.

Figure 7b represents data for the configuration in which the two-per-rev cam covered the entire nozzle control area. Data were taken for the cam in proximity to the nozzle with nominal clearances of 0.01 and 0.26 in. at the minimum gap and with a tapered slot distribution in the pipe. The 0.26-in. spacing had the same effect with the two- as with the one-per-rev, i.e., the loss coefficient was reduced as the nozzle control area increased. The change in slot distribution had no noticeable effect on the loss coefficient.

Data for the one-per-rev cam in proximity to the nozzle with a nominal clearance of 0.51 in. at the minimum gap showed no indication of pressure or mass flow rate modulation. These data serve as a loss coefficient versus pressure calibration for the nozzle-slot combination; see Figure 8. This curve is the limiting loss coefficient for the curves in Figure 7. The loss coefficient as defined above and presented in Figure 7 was used in a computer program to design the cam valve system for the higher harmonic circulation control (HHCC) rotor model which was successfully built, tested, and evaluated at NSRDC, Carderock.

EFFECT OF VARIOUS CONFIGURATIONS ON TOTAL PRESSURE

A question foremost in the mind of a user of this type of flow control system concerns the uniformity and controllability of the output total pressure wave when the configuration changes from all one-per-rev to all two-per-rev while the plenum pressure remains constant. The visual solution presented in Figure 9 shows the uniformity and the relative strength of each component. The two cycles of total pressure data with a plenum pressure of 8.84 psi and a frequency of 16.7 Hz indicate that there was a smooth transfer of the total pressure wave from a two- to a one-per-rev component as the percentage of two-per-rev cam input decreased from 100 to 0 percent. Conversely, the percentage for the one-per-rev cam increased from 0 to 100 percent.

Figure 9 includes the data for a configuration where there are equal proportions of the nozzle height covered by the one-per-rev, the two-per-rev, and a constant area. Note that the mean value of the pressure curve increased due to the constant area but that strong one-per-rev and two-per-rev components were maintained. The data presented in Figure 9 are typical of the data at other pressures and frequencies.

When the cam system is applied to a circulation control helicopter rotor blade, it is important to know whether the cam configuration affects the average pressure for various radial positions. The average total pressure of a particular pipe station was normalized by the plenum pressure and is plotted in Figure 10 versus the percent of pipe length x/ℓ. These data are for a limited number of configurations which include the 100- and 67-percent pipe volumes and the nozzle control area. These data are for different frequencies provided by the one- and the two-per-rev cams.

Plenum pressures of 13.75 and 2.95 psi were used in conjunction with the one-per-rev cam configuration to determine any effect of plenum pressure on total pressure losses within the pipe. Since no appreciable effect of plenum pressure on the pipe total pressure losses was found, the one-per-rev data for the 2.95-psi case were omitted for clarity.

The nozzle control area is the same as defined earlier in this report. However, the data presented in Figure 10 are for rotating cams, and the areas cannot be compared directly by the gap (the nominal clearance plus the cam eccentricity) times the nozzle periphery. The total integrated area around the azimuth gives a more meaningful area representation. The various configurations are presented as percentages, with the one-per-rev cam integrated area designated as a reference whose magnitude is 3.195 in.^{2}. A comparison of the one- and two-per-rev cam data shows a significant reduction in the average pipe total pressure due to a decrease in integrated nozzle control area of approximately 58 percent; see Curves A and B in Figure 10. This reduction in the integrated nozzle area was caused by the differences in eccentricity of the one- and two-per-rev cams.

It was stated earlier that changing the plenum pressure from 2.95 to 13.75 psi had no noticeable effect on the pipe total pressure at a particular station. This enables Curves A and C to be compared directly. The addition of a 0.25-in. spacer increased the reference integrated nozzle control area by approximately 320 percent and thereby increased the total pressure ratio by approximately 250 percent. As the control area increased, the mass flow also increased and caused higher velocities inside the pipe. The higher velocities caused

larger frictional losses; these resulted in total pressure losses as x/ℓ increased. This pressure loss is indicated in Curve C of Figure 10 which shows a 10-percent loss in total pressure ratio. For the smallest control area (Curve B) which had a much smaller mass flow, there was no reduction in pipe total pressure as x/ℓ increased.

The effect of the pipe volume on the total pressure can be observed by comparing Curves A and D of Figure 10. For a given plenum pressure, the reduced pipe volume (67 percent) had an average overall increase in total pressure of 15 percent. This increase was partially due to the rearranging of the streamlines within the pipe when the insert was installed, but it was mainly caused by the reduced volume within the pipe (this resulted in high internal velocities). As discussed above, when x/ℓ was increased for the reduced-volume configuration the higher velocities caused a loss in total pressure of approximately 11 percent.

EFFECT OF VARIOUS CONFIGURATIONS ON PEAK-TO-PEAK PRESSURE

The application of circulation control to a helicopter rotor requires that the internal blade pressure and mass flow be cycled in a specified manner and that they be maintained at some constant level. The rotor trim is provided by the cyclic pressure and mass flow variation, and the mean lift is provided by the constant pressure. In evaluating or determining the usability of this type of cam valve system, it is important to know the effect of the nozzle control area and the reduced pipe volume on the peak-to-peak (cyclic) pipe entrance total pressure. To demonstrate the effect for the various configurations tested, the peak-to-peak entrance total pressure is plotted in Figure 11 versus the plenum pressure for different frequencies provided by the one- and two-per-rev cams. A comparison of the nominal clearance (0.01-in.) configuration and the 0.25-in. spacer configuration enables a rough estimate of the sensitivity of the radial position of the cams with respect to the nozzle. The data for these configurations are presented as Curves A and B of Figure 11. The method used to increase the gap between the cam and the nozzle raised the mean value of the total pressure; however, only the peak-to-peak value is discussed here. As shown in Figure 11, the peak-to-peak total pressure at the entrance of the pipe was dependent on the relative frequency of rotation of the cam with respect to the nozzle. Curve A for the configuration of the one-per-rev cam was arbitrarily designated as the reference integrated nozzle control area as discussed previously. The one-per-rev cam with a 0.25-in. additional gap (Curve B) increased the reference integrated nozzle control area by 320 percent.

Some general trends and observations can be noted from Figure 11. For a plenum pressure of about 7 psi, the peak-to-peak pipe entrance total pressure for the reference integrated nozzle control area (the one-per-rev cam configuration) was approximately three times the comparable value for an integrated area increase of 320 percent. This is a relatively large reduction in the peak-to-peak pressure for a configuration where the gap introduced between the cam and the nozzle was approximately equal to the maximum eccentricity of the cam. The two-per-rev configuration represents a 58-percent reduction in reference integrated nozzle control area. These data are presented as Curve C of Figure 11. A comparison of Curves A and C shows the effect of reducing the eccentricity of the cam by 34 percent, namely, a reduction of the peak-to-peak entrance total pressure by a factor of approximately three.

Curves C and D of Figure 11 show the effect of the tapered slot on the peak-to-peak inlet pressure. The effect of reducing the volume of the pipe by 67 percent may be seen by comparing Curves A and E.

Figure 12 presents the peak-to-peak total pressure versus the rotational frequency for the middle and end stations of the pipe. The data have been normalized by the peak-to-peak entrance total pressure. The data for the one-per-rev cam (Figure 12a) include the three basic configurations and the effect of an increase in the plenum pressure (very large differences in peak-to-peak pressure characteristics). The data for the two-per-rev cam (Figure 12b) include the effect of a tapered slot distribution and the effect of a smaller plenum pressure difference. The higher frequency represented by the two-per-rev cam showed a much larger peak-to-peak pressure at the end station than at the middle station; see Figure 12b.

DISCRETE AZIMUTH POSITION DATA

Figure 13 gives a comparison of data collected by the discrete azimuth position method and by rotation of the cam at some prescribed frequency. As indicated in Figure 13a (one-per-rev cam) the agreement between the two methods was very good at a frequency of 49 Hz. This agreement was observed for other frequencies and was further verified by data for the two-per-rev cam configuration (Figure 13b). This verification enables the discrete azimuth position method to be used to obtain design data and to evaluate the performance of a rotating cam valve system such as described in this report. Figure 14 presents the static and total pressure versus azimuthal position for various additional configurations.

Figure 15 shows the air mass flow rate versus the azimuth position for the various configurations tested. These data were obtained by the discrete azimuth positioning technique described above and are presented to verify the fact that for various configurations and plenum pressures, the cams will give a mass flow rate regulation in agreement with their inputs. Also included is the average mass flow rate from the rotating cam data.

EFFECT OF VARIOUS CONFIGURATIONS ON PEAK-TO-PEAK STATIC AND TOTAL PRESSURE

Figure 16 indicates the peak-to-peak pipe static and total pressures versus rotational frequency for various configurations and plenum pressures. Here again, only the peak-to-peak pressure value is presented. An estimate of the mean value for the one-per-rev configuration requires only that the peak-to-peak value be divided by two. However, this is not true for the two-per-rev cam configuration because of its nonuniformity of construction. The one-per-rev cam data indicate the following general trends: (1) as the plenum pressure was increased, the peak-to-peak pressures were attentuated for increasing frequencies and (2) the peak-to-peak pressure was higher for a 33-percent reduction in pipe volume but exhibited the same general characteristics with increasing frequencies as for the 100-percent pipe volume; compare Figures 16a and 16b.

The general trends are not as clear for the two-per-rev configurations. For the higher plenum pressures, the peak-to-peak pressures were attentuated up to a particular frequency and were amplified above it. For the lower plenum pressure, the peak-to-peak pressures were amplified for all frequencies tested;

see Figure 16c. Slot distribution raised the peak-to-peak pressure on the two-per-rev configuration but did not affect the generalizations stated above; compare Figures 16c and 16d. The 0.25-in. spacer on the one- and two-per-rev configurations raised the pipe entrance total pressure peak-to-peak value well above those observed for the configuration without a spacer. No strong frequency effect was observed for the 0.25-in. spacer configurations (Figure 16e).

Because it is quite simple, this mockup did not allow the determination of several important factors. The phase shift in the amplifiers precluded evaluating the time phase shift in the pressure between the plenum, pipe entrance, and pipe end. Moreover the instrumentation limitation made it impossible to get a breakdown of the individual effect of each component, i.e., the penalties involved in combining various combinations of constant and one- and two-per-rev components. These shortcomings of the investigation were also a result of the poorly manufactured cams which resulted in waveforms other than sinusoidual. The thicknesses of the spacers used to increase the gap between the cam and nozzle were arbitrarily chosen to be 0.25 and 0.50 in. These did not allow a detailed evaluation of the pressure and mass flow since the nozzle is removed radially from the cam.

CONCLUSIONS

A preliminary investigation was performed to determine the effect of some basic configurations on the modulation of the pressure and the mass flow rate within a round receiver. Several conclusions have been drawn:

1. Air pressure and mass flow rate can be modulated by means of a simple cam valve system.

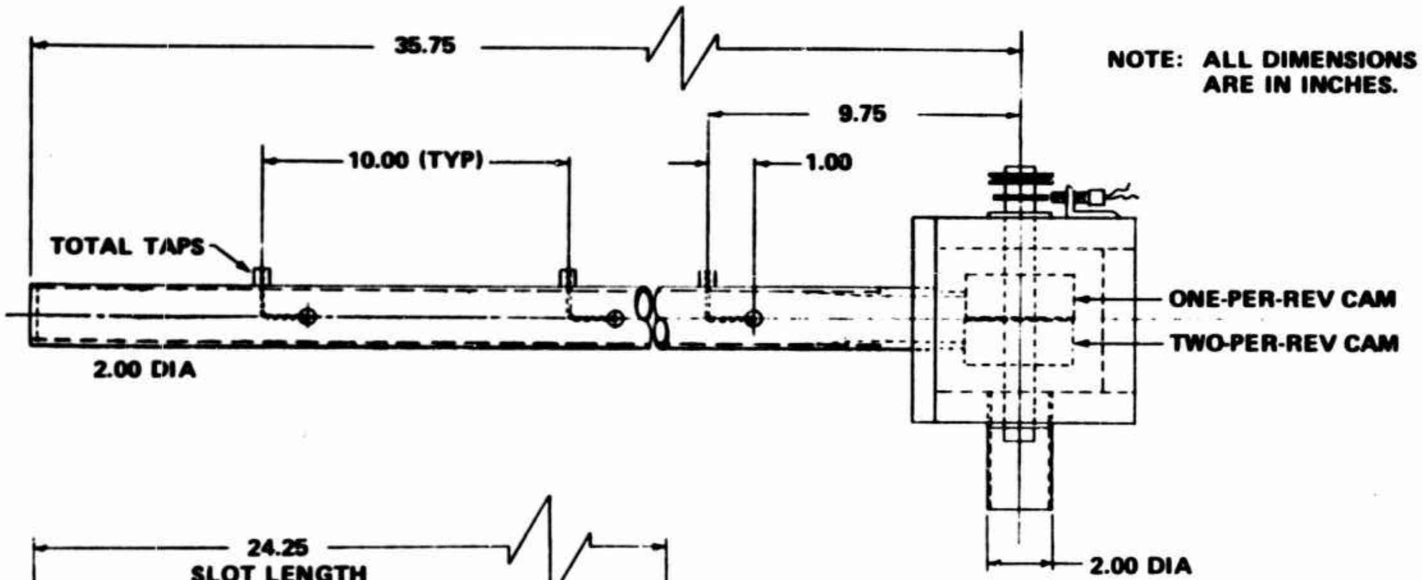
2. As the gap between the periphery of the cams and the nozzle increased for a given cam geometry, the mean pressure and mean mass flow rate increased, whereas the peak-to-peak pressure and peak-to-peak mass flow rate decreased. As the gap was increased from 0.26 to 0.51 in., the ability of the valve system to modulate the pressure and mass flow rate ceased.

3. There was a smooth transfer of the total pressure and mass flow rate from a two- to a one-per-rev component (or vice versa) as the percentage of two-per-rev cam input was decreased from 100 to 0 percent.

4. The change in slot height distribution had no noticeable effect on the loss coefficient but the average mass flow was reduced by approximately 15 percent.

5. When the receiver volume was decreased by 33 percent, (a) the average mass flow decreased by approximately 2 to 5 percent, (b) the loss coefficient was considerably reduced, (c) the average overall total pressure was increased by 15 percent, and (d) the peak-to-peak pressure was increased and the same general characteristic was maintained with increasing frequency.

Figure 1 – Details of the Mockup Control Valve



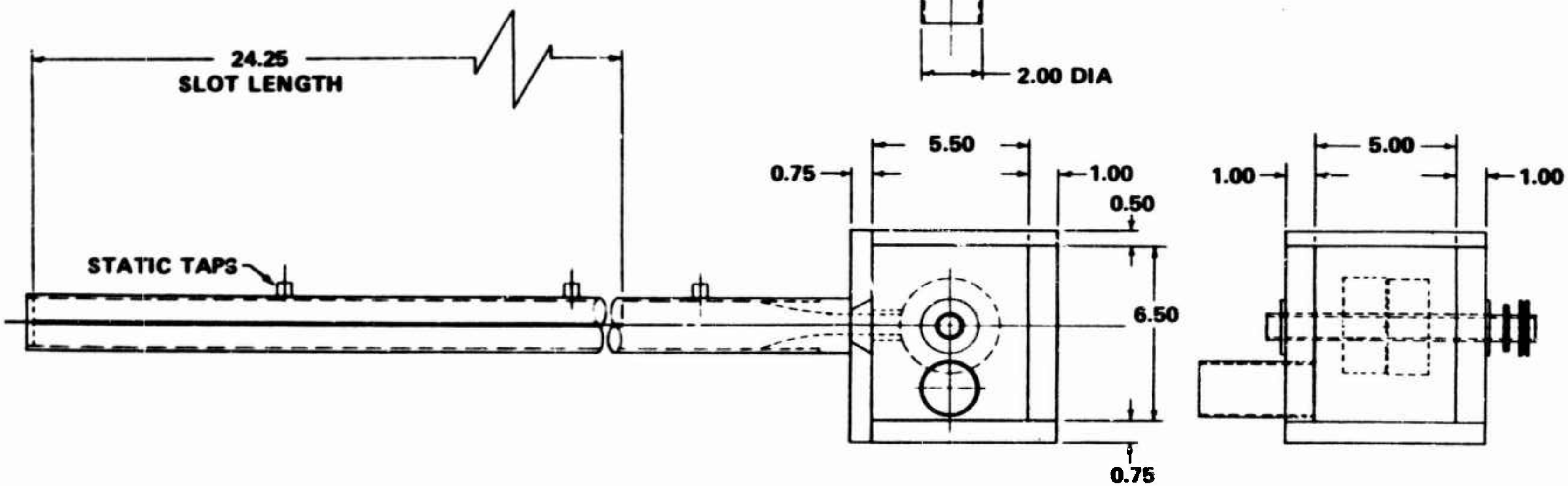
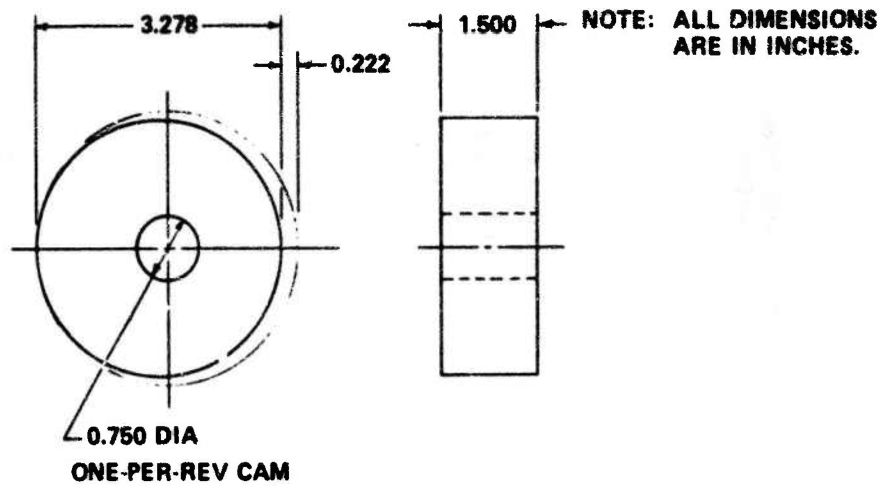
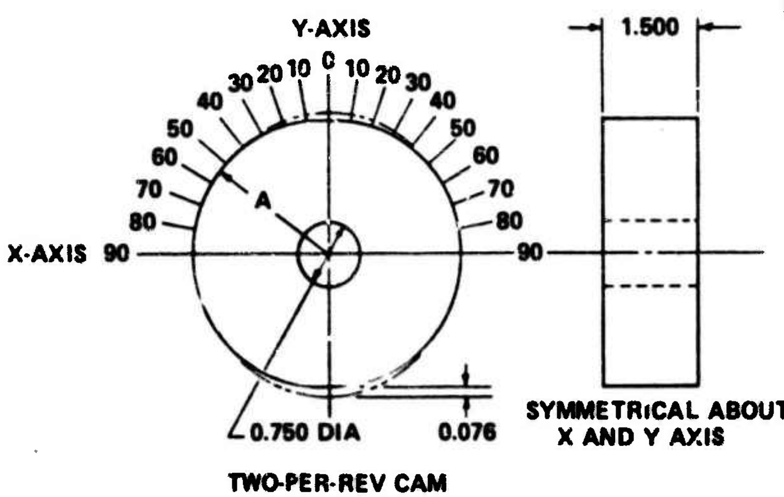


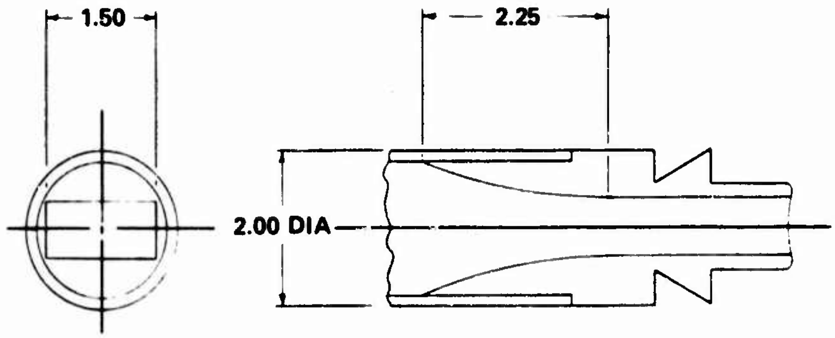
Figure 1a – General Arrangement and Principal Dimensions of the Model





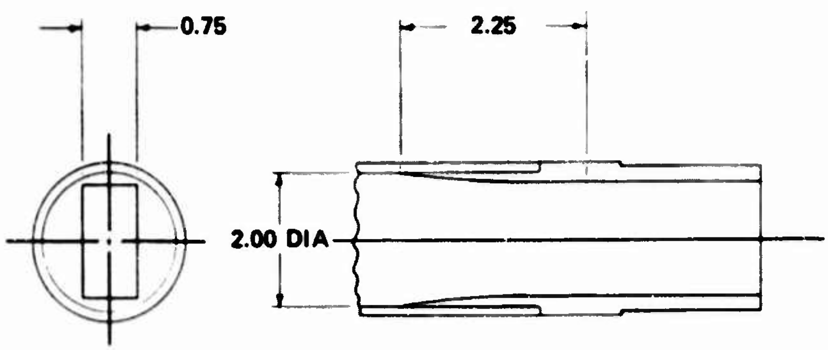
ANGLE DEGREES	DIM A (RADIUS)
0	1.6740
10	1.6763
20	1.6829
30	1.6930
40	1.7054
50	1.7186
60	1.7310
70	1.7411
80	1.7477
90	1.7500

Figure 1b – One- and Two-per-Rev Cam Details



TOP VIEW

NOTE: ALL DIMENSIONS
ARE IN INCHES.



SIDE VIEW

Figure 1c – Nozzle Details

Figure 2a – Hub Valve Test Setup

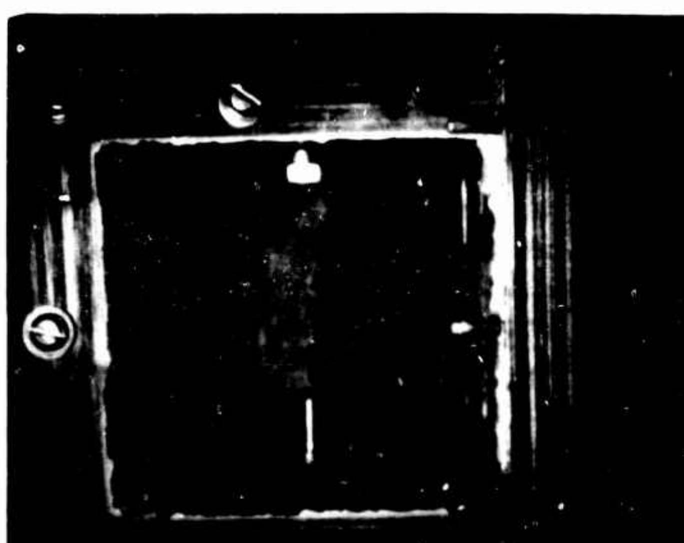


Figure 2b – Closeup View of One- and Two-per-Rev Cams

Figure 2 – Experimental Arrangement

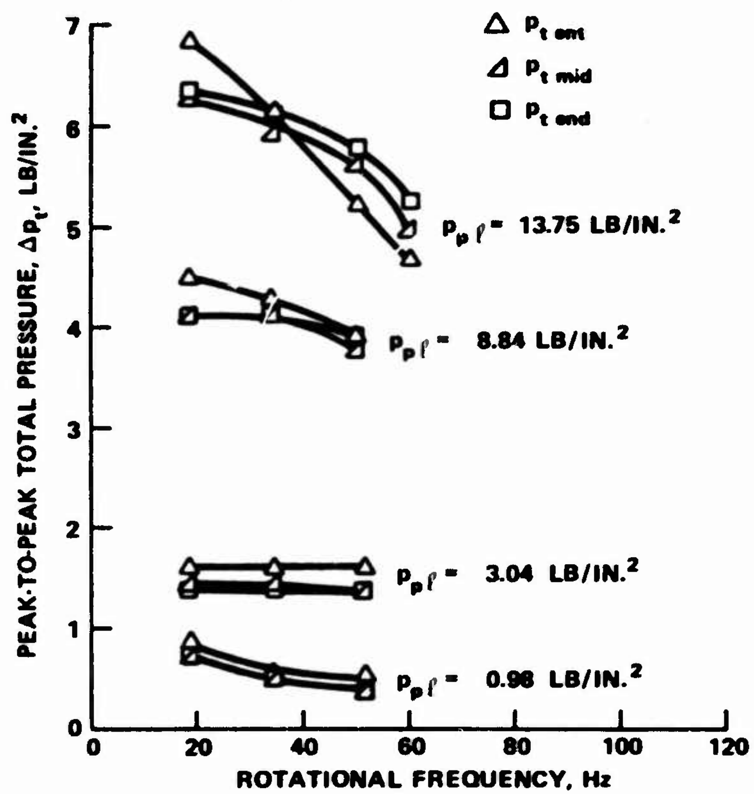


Figure 3a – One-per-Rev Cam Configuration

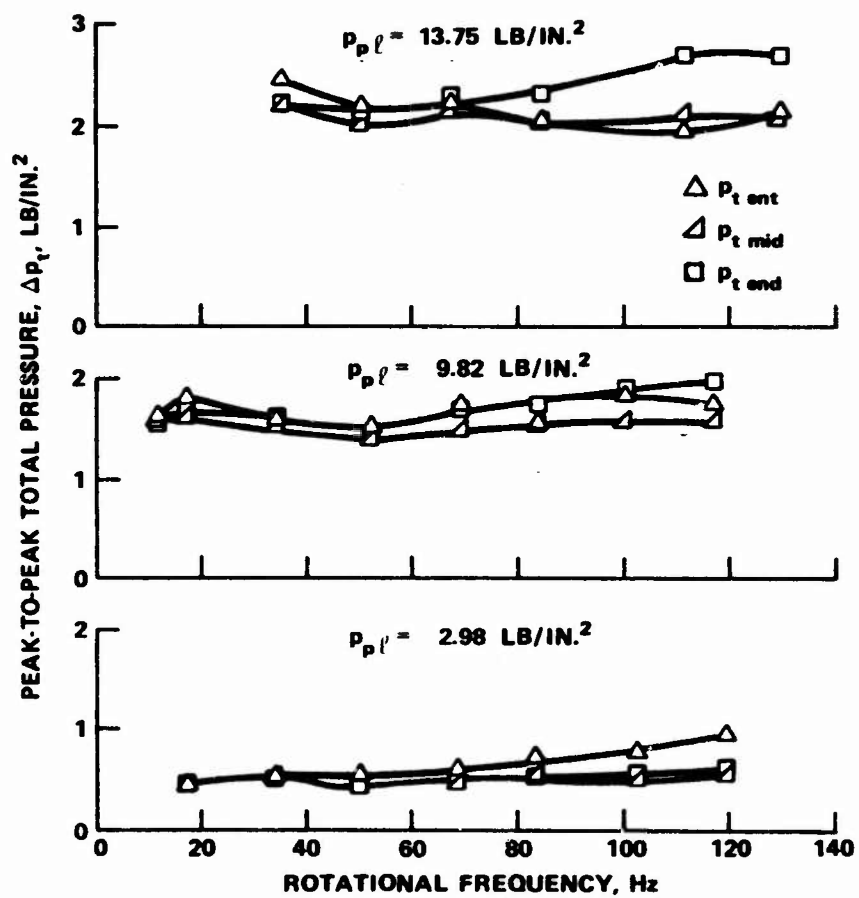


Figure 3b – Two-per-Rev Cam Configuration

Figure 3 – Effect of Rotating Frequency on the Peak-to-Peak Total Pressure

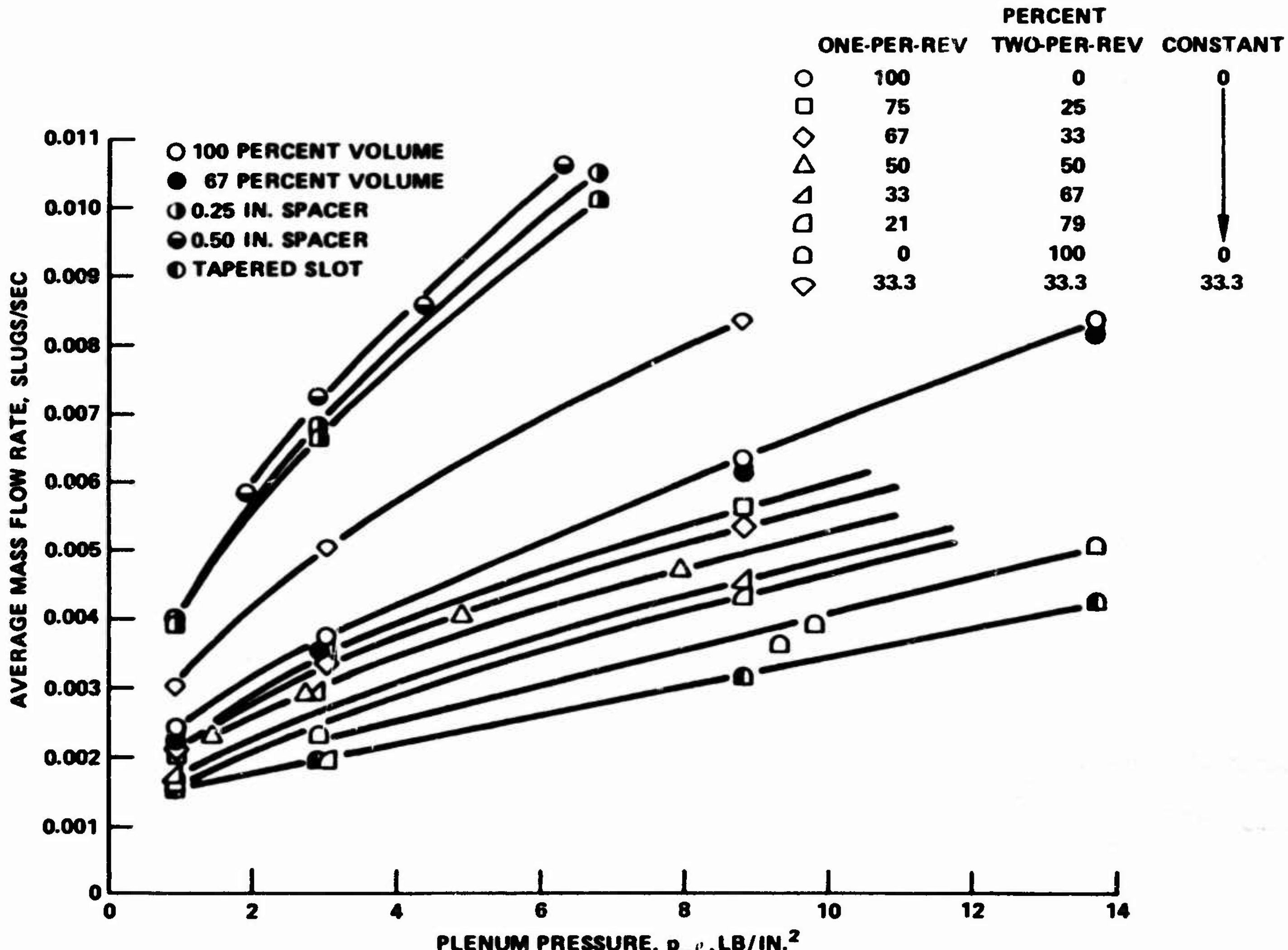


Figure 4 – Average Mass Flow Rate versus Plenum Pressure

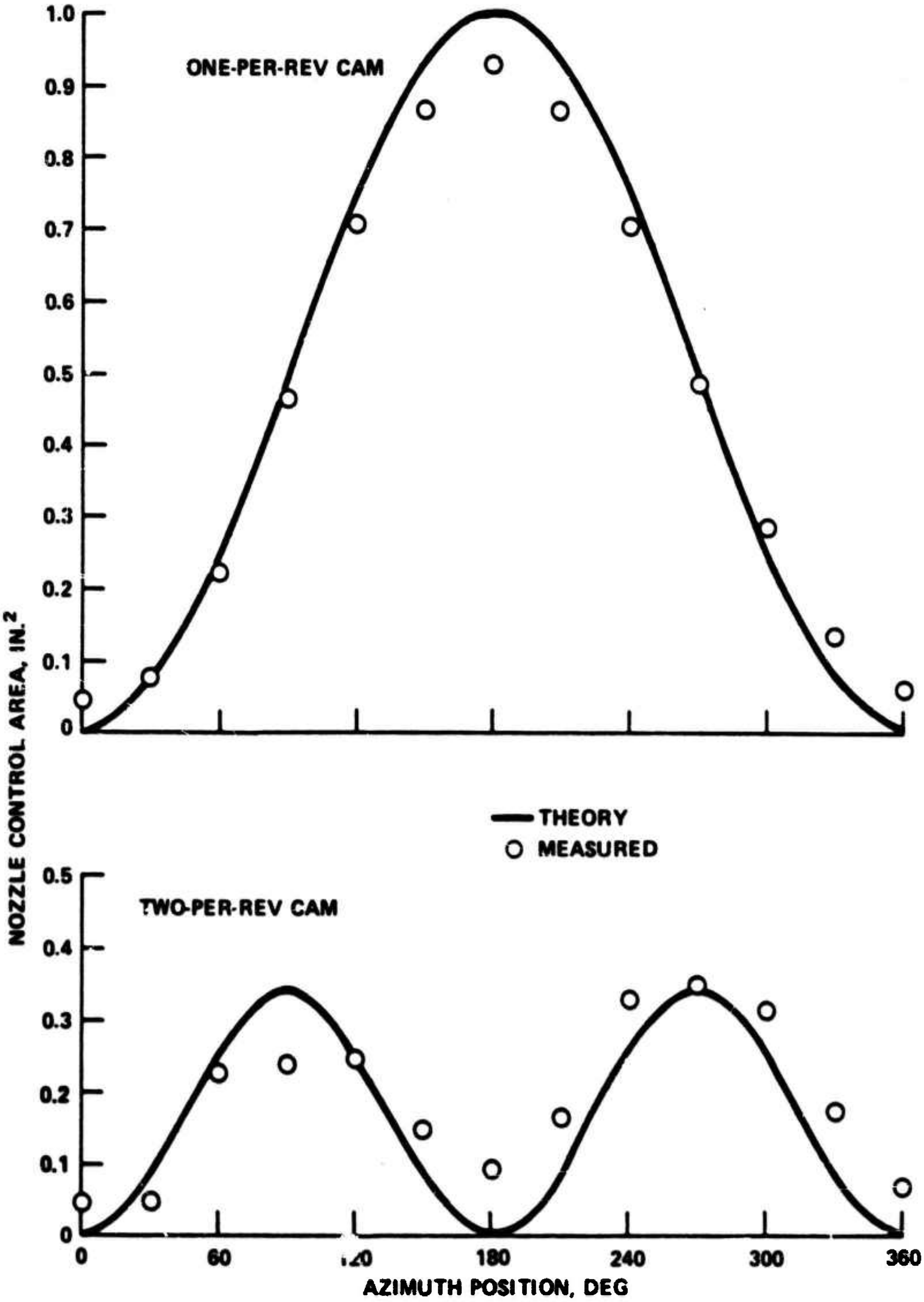


Figure 5 – Nozzle Control Area versus Azimuth Position

Figure 6 – Mass Flow Rate versus Nozzle Control Area

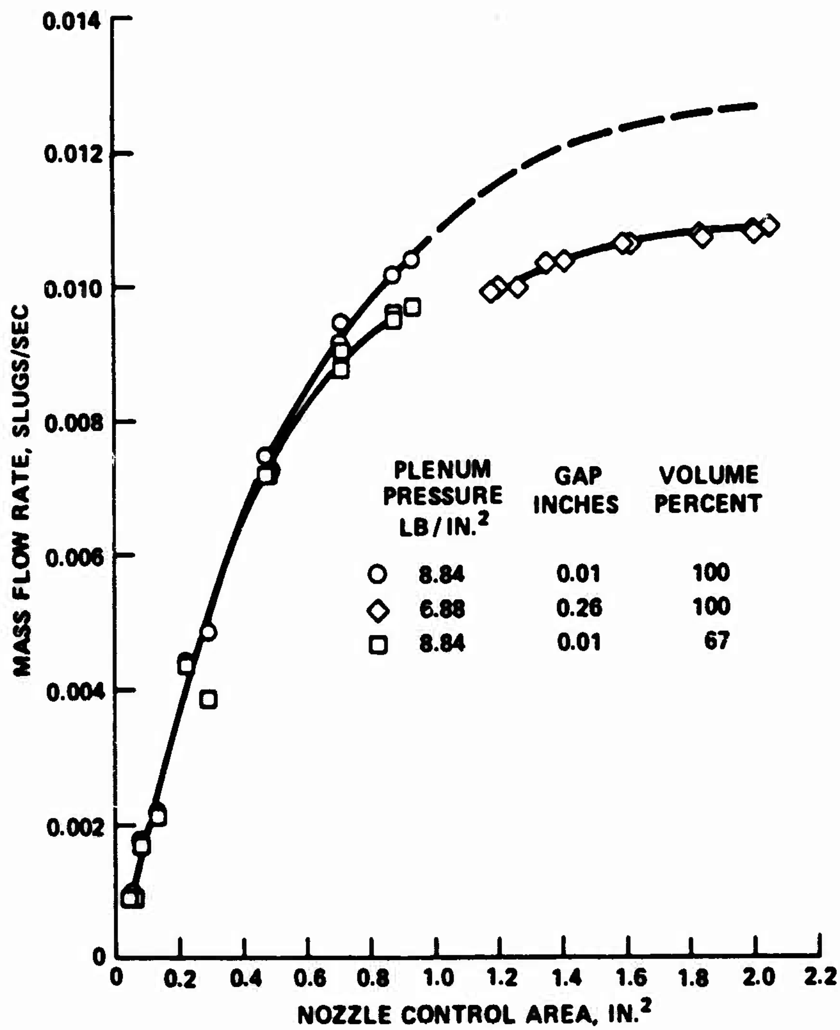


Figure 6a – One-per-Rev Cam Configuration

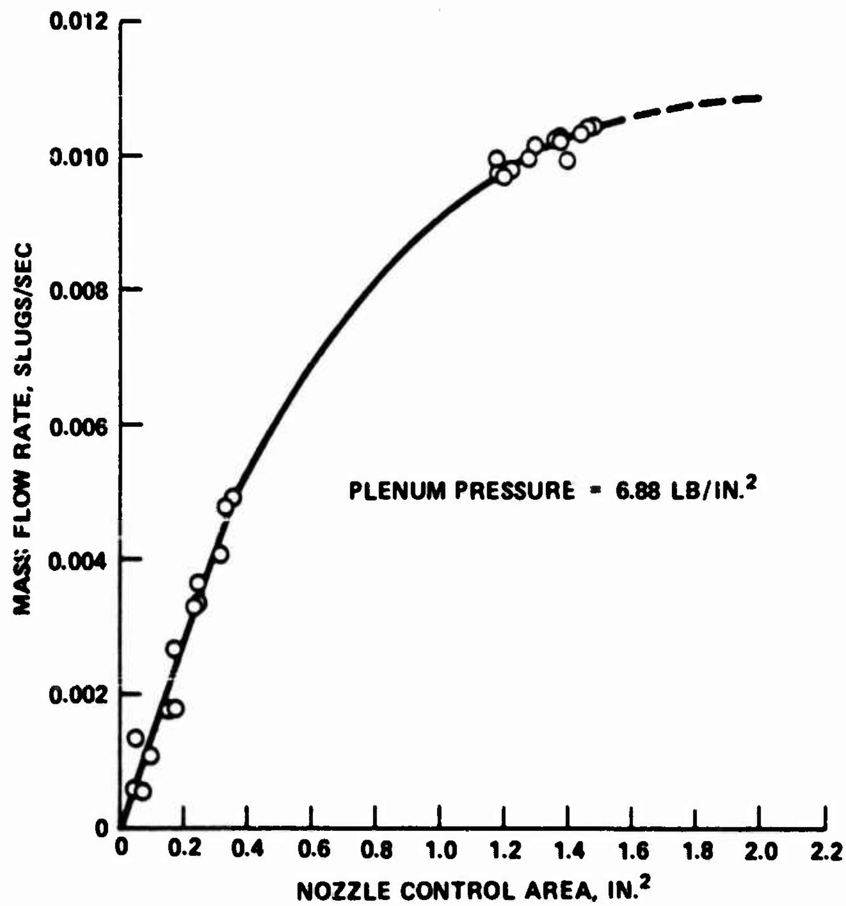


Figure 6b – Two-per-Rev Cam Configuration

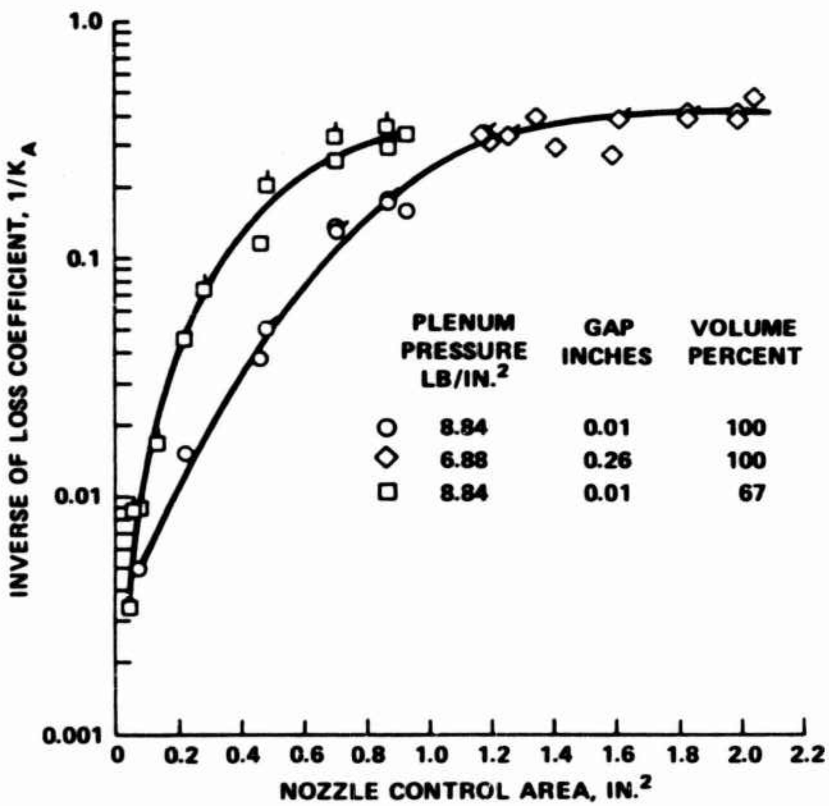


Figure 7a – One-per-Rev Cam Configuration

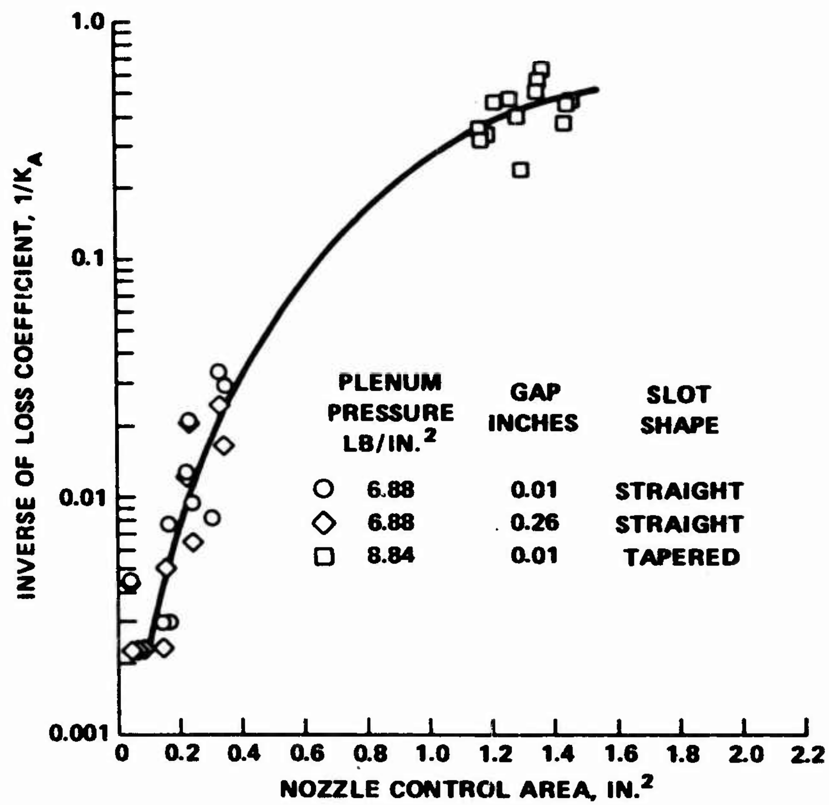


Figure 7b – Two-per-Rev Cam Configuration

Figure 7 – Inverse of the Loss Coefficient versus Nozzle Control Area

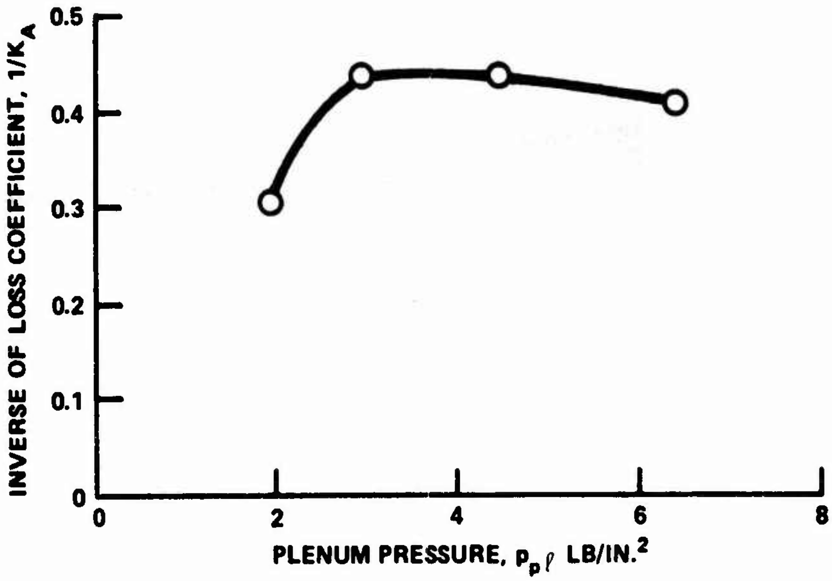


Figure 8 – Inverse of the Loss Coefficient versus Plenum Pressure for the Nozzle without the Cams

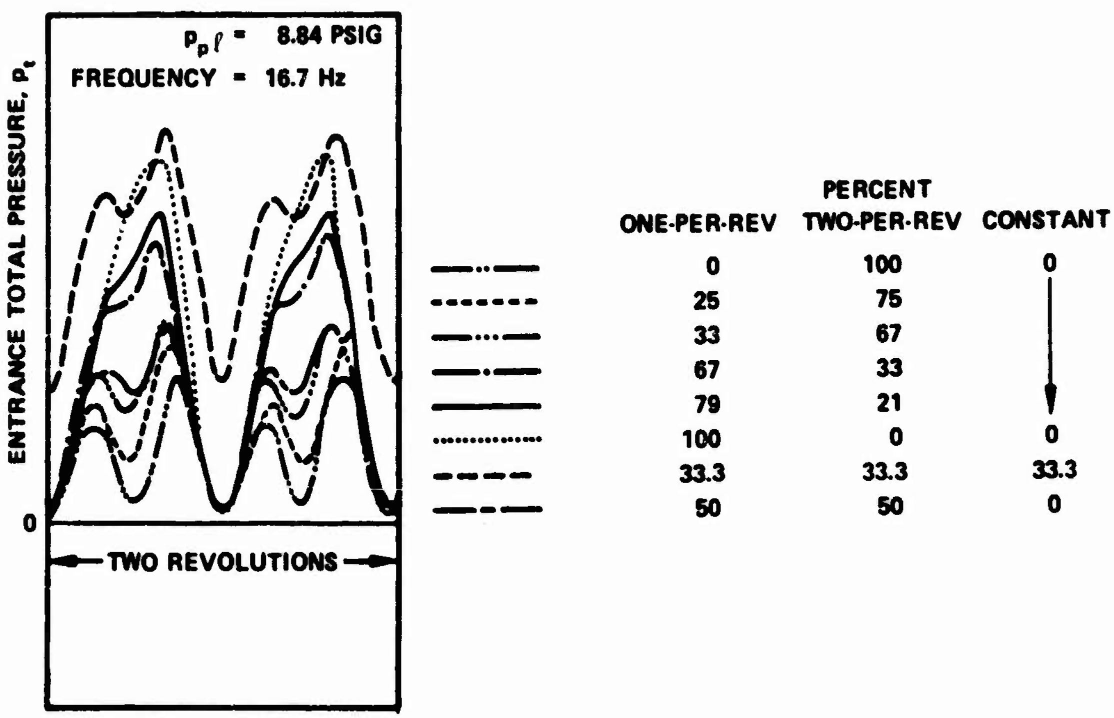


Figure 9 – Total Pressure Variation for Various Combinations of Constant, One-per-Rev and Two-per-Rev Components

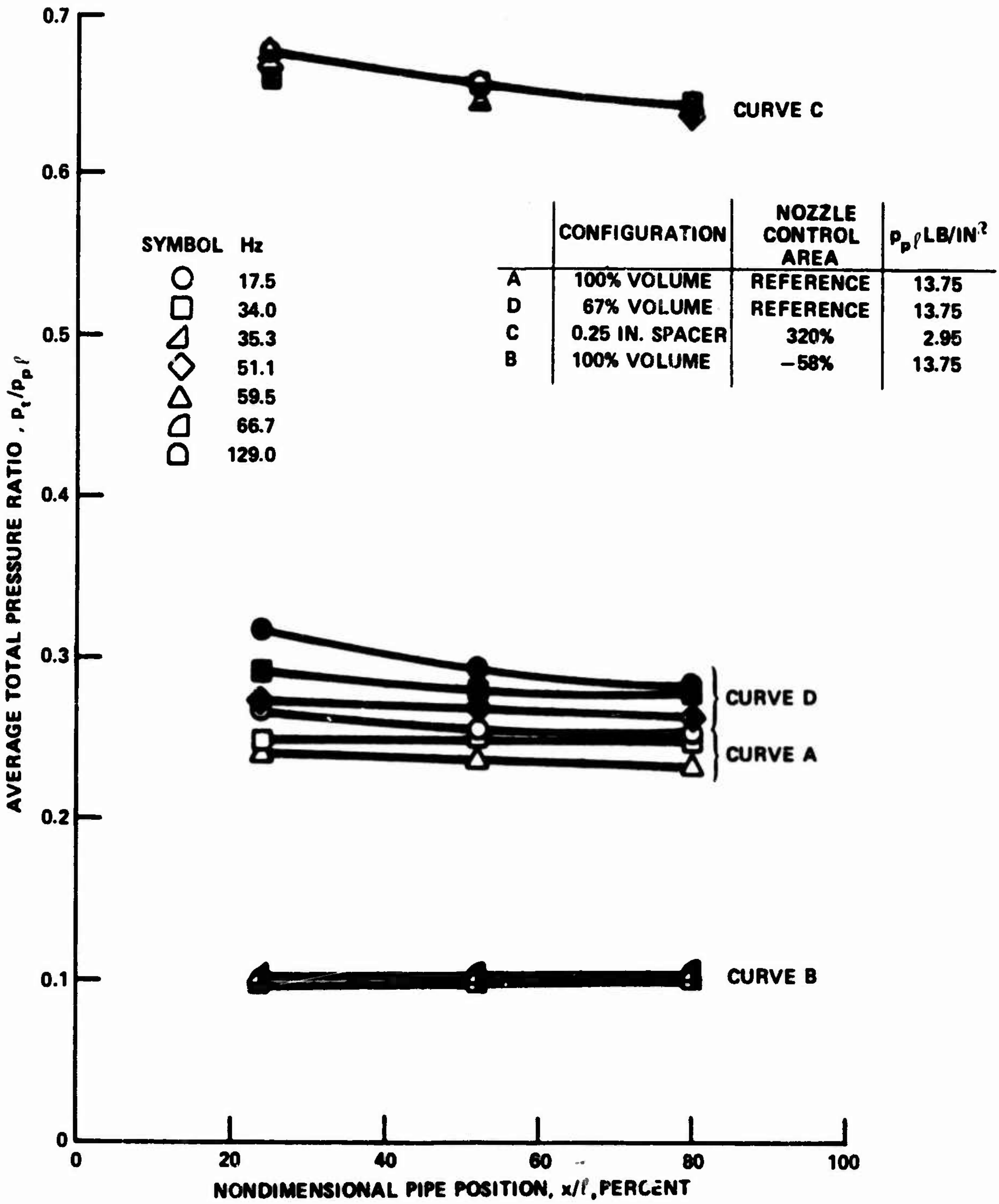


Figure 10 – Average Total Pressure Ratio versus Percent Pipe Length

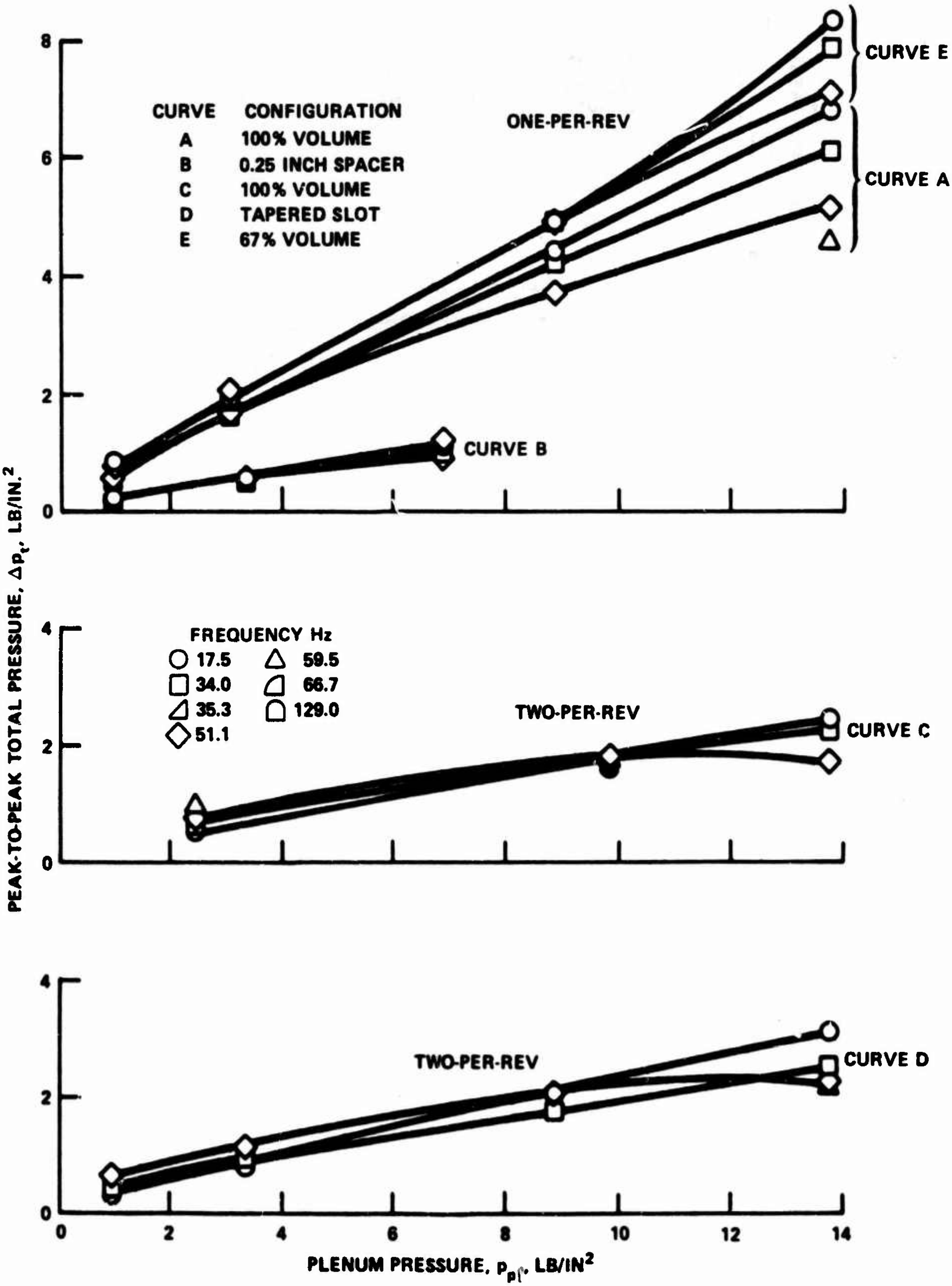


Figure 11 – Peak-to-Peak Entrance Total Pressure versus Plenum Pressure

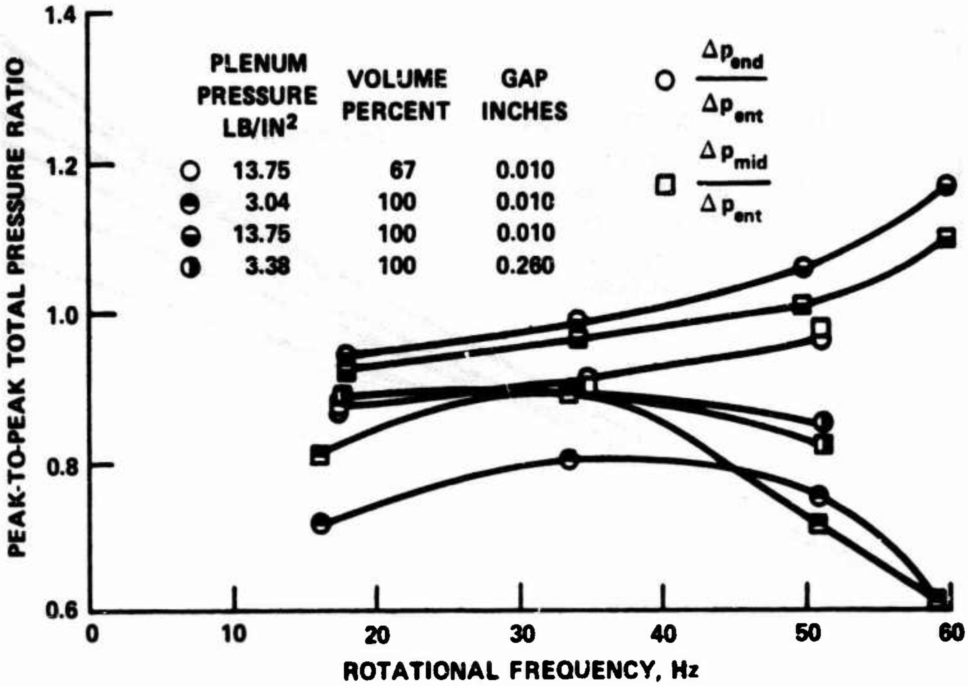


Figure 12a – One-per-Rev Cam Configuration

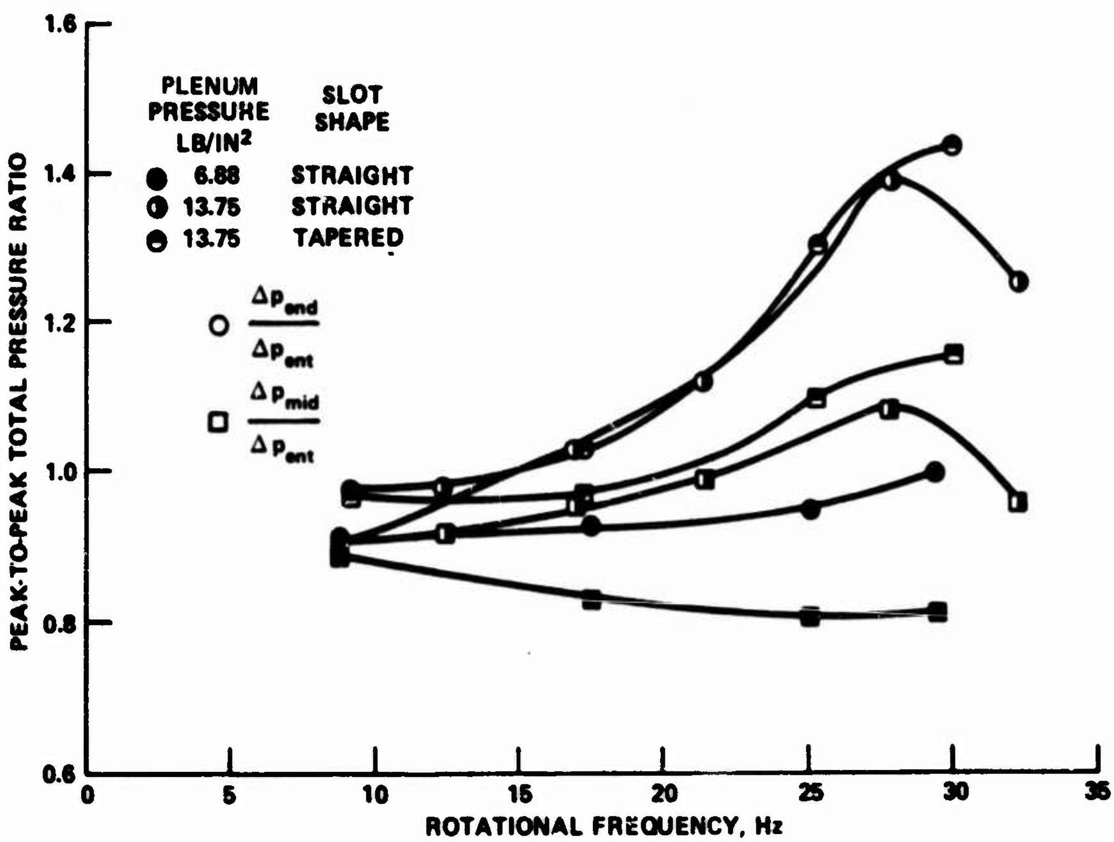
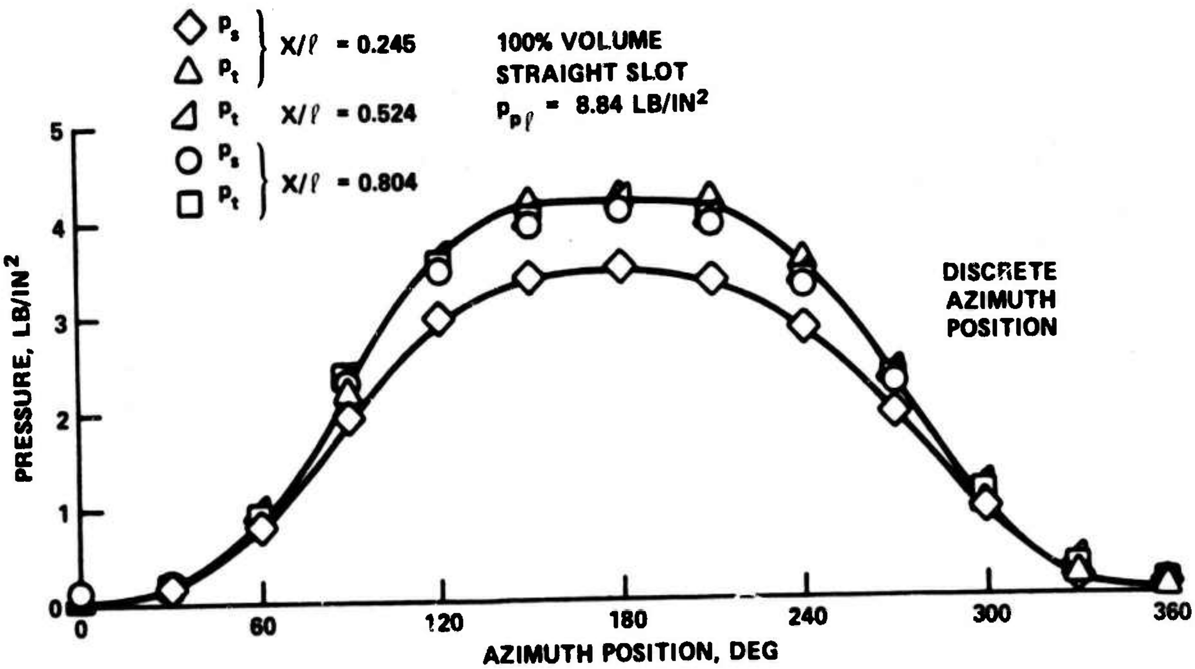


Figure 12b – Two-per-Rev Cam Configuration

Figure 12 – Peak-to-Peak Total Pressure Ratio versus Rotational Frequency

Figure 13 – Comparison of Discrete Azimuth Position Data with Dynamic Data



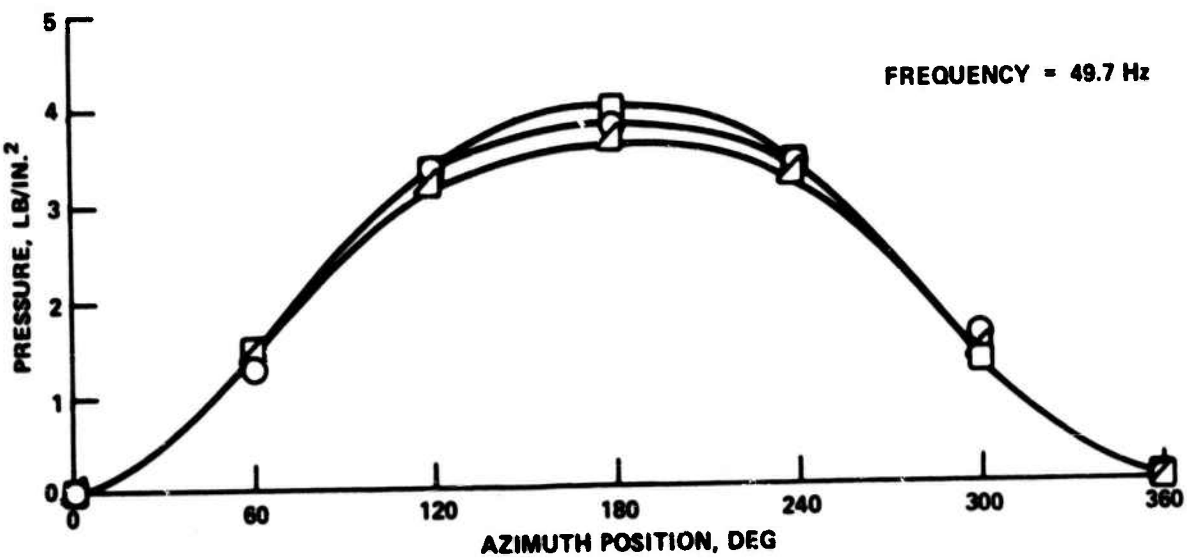


Figure 13a – One-per-Rev Cam Configuration

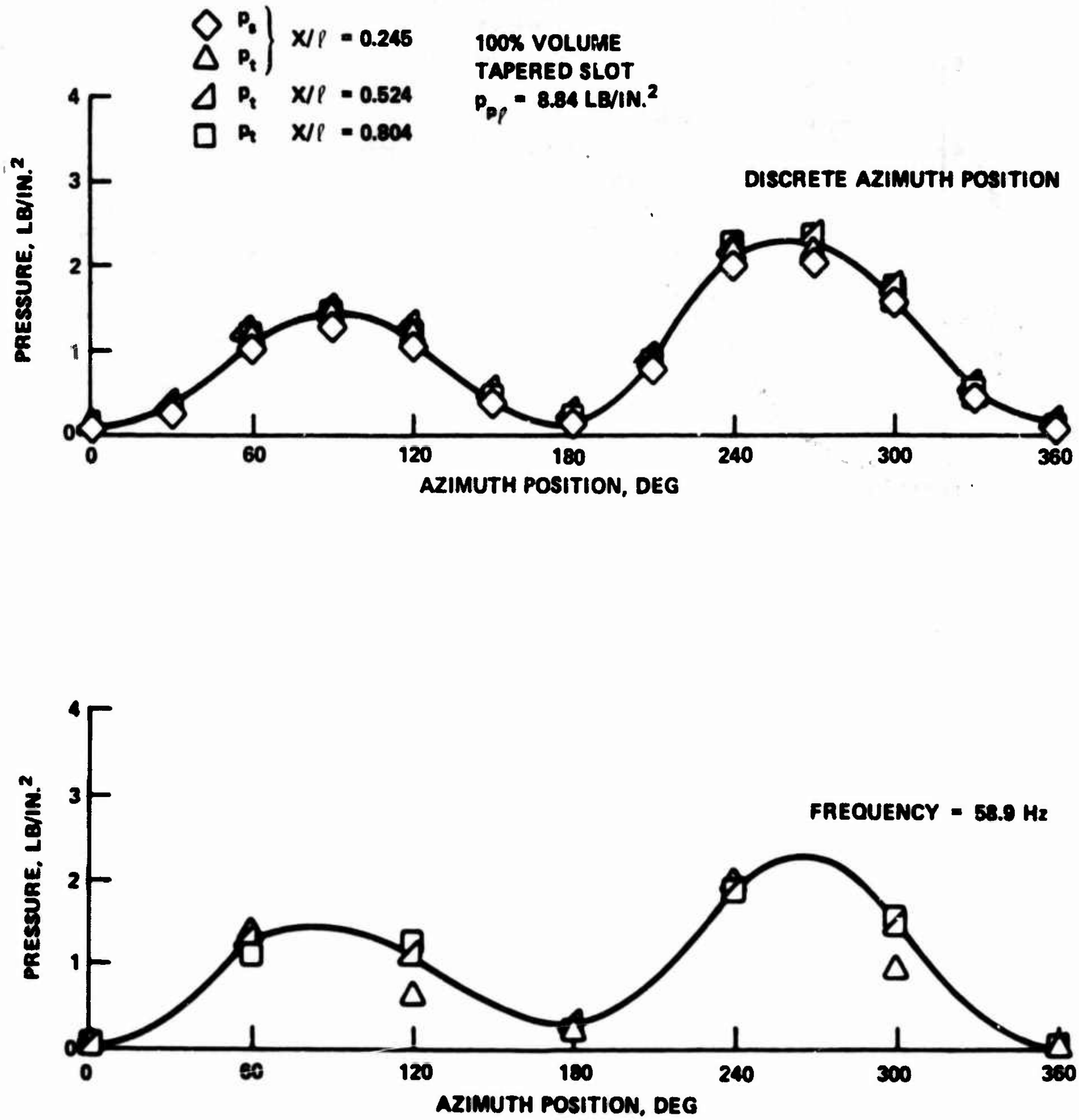
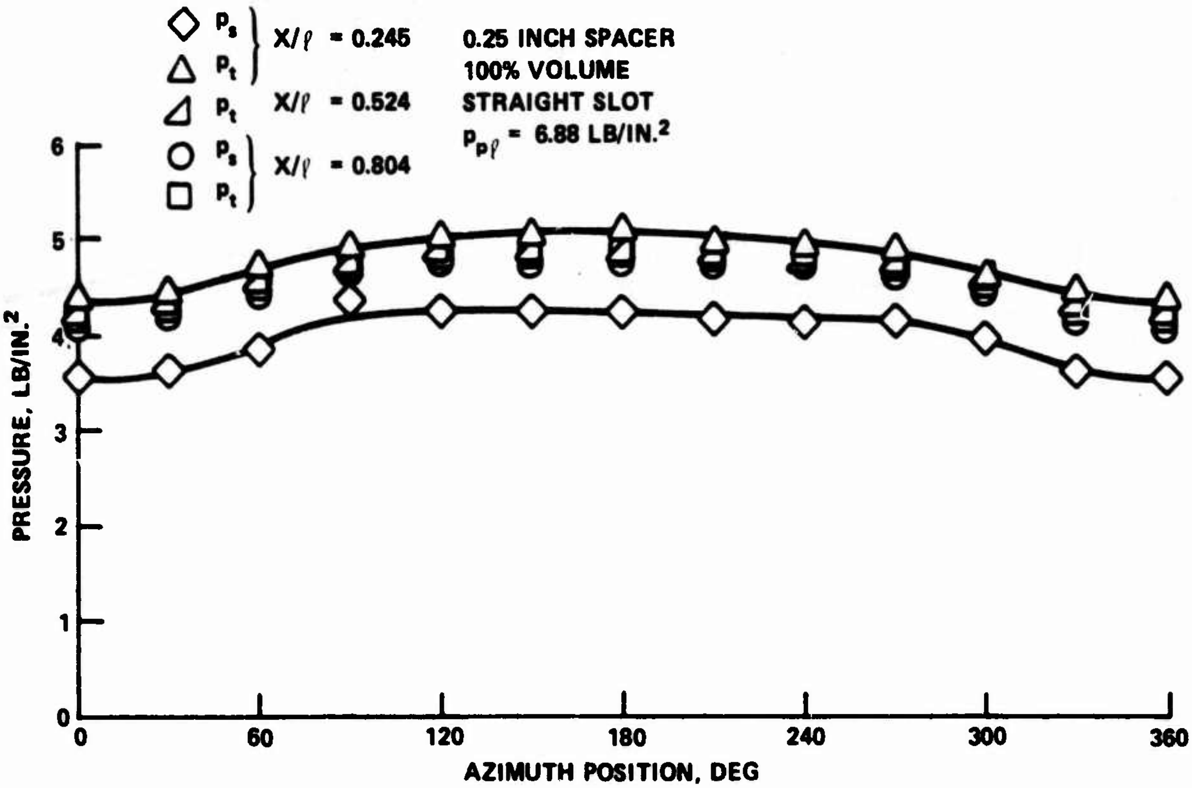


Figure 13b – Two-per-Rev Cam Configuration

Figure 14 – Discrete Azimuth Position Data for Additional Configurations



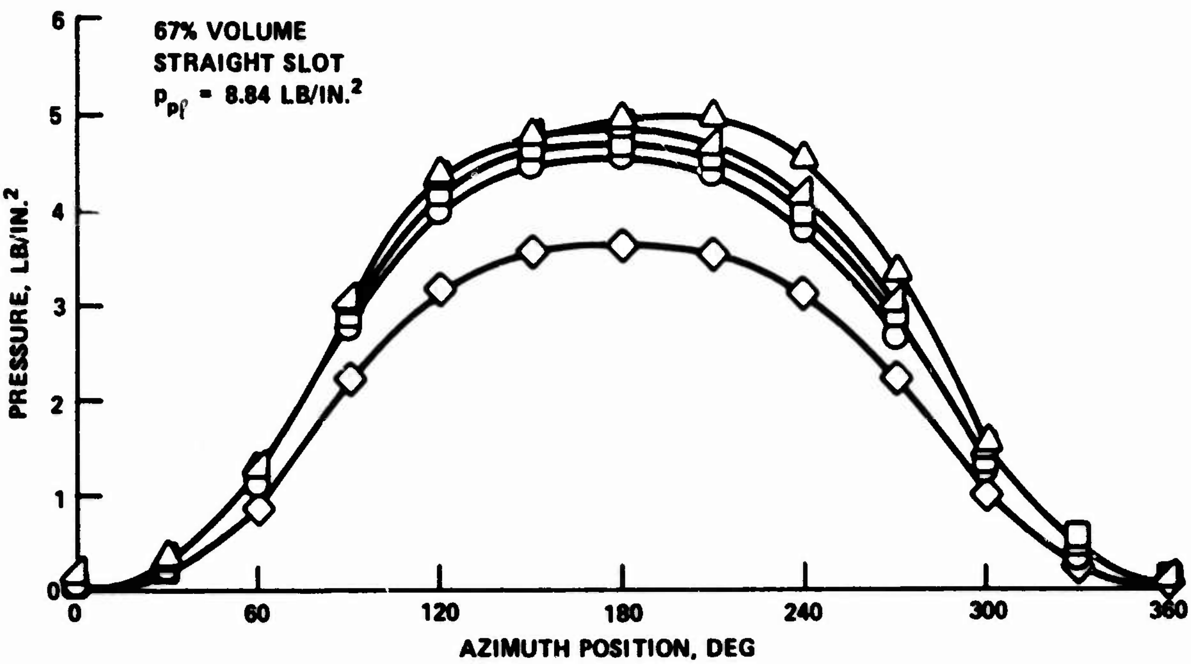
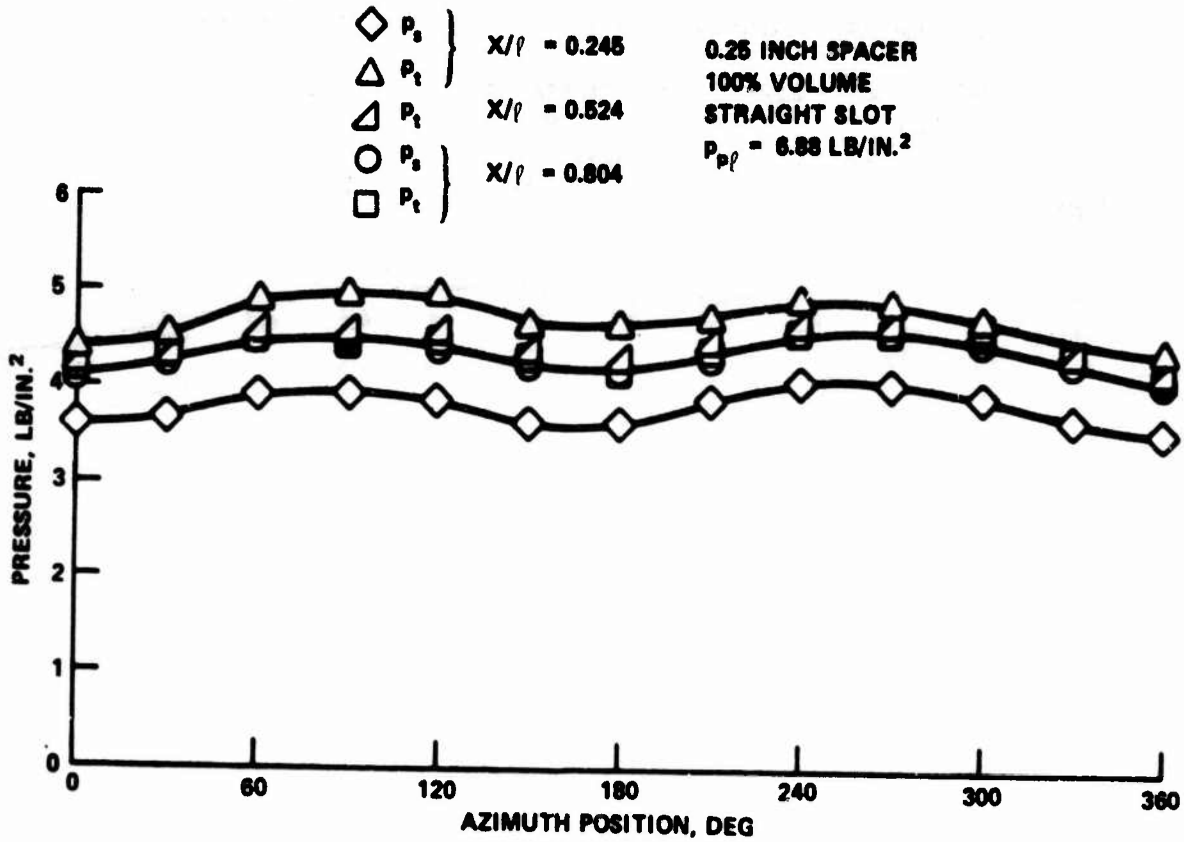


Figure 14a – One-per-Rev Cam Configuration



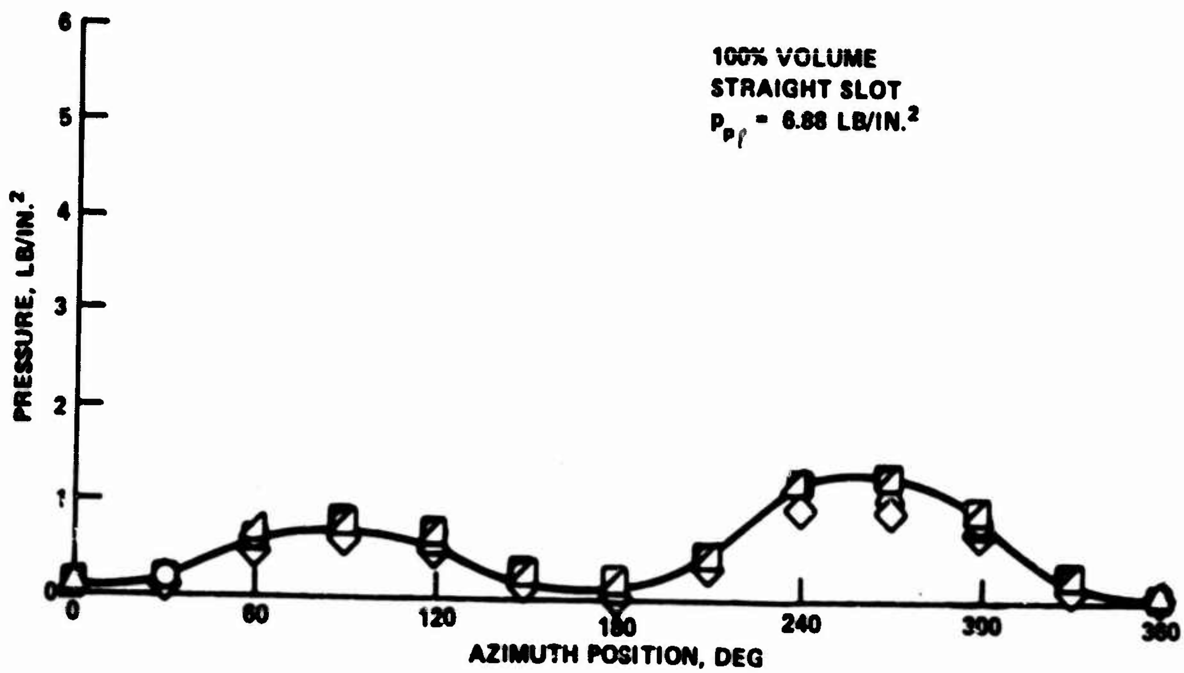


Figure 14b – Two-per-Rev Cam Configuration

Figure 15 – Mass Flow Rate versus Azimuth Position for Various Configurations

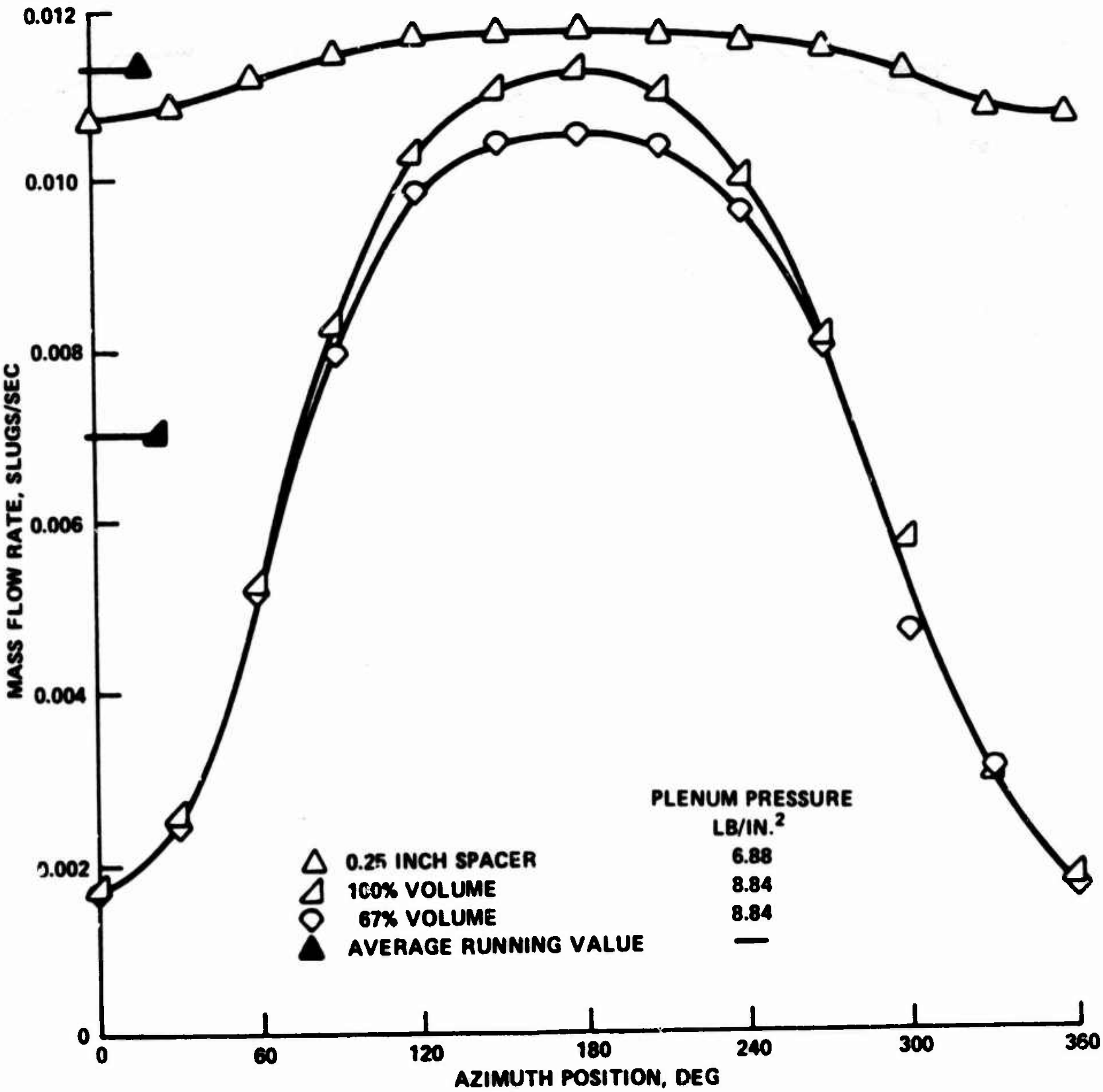


Figure 15a – One-per-Rev Cam Configuration

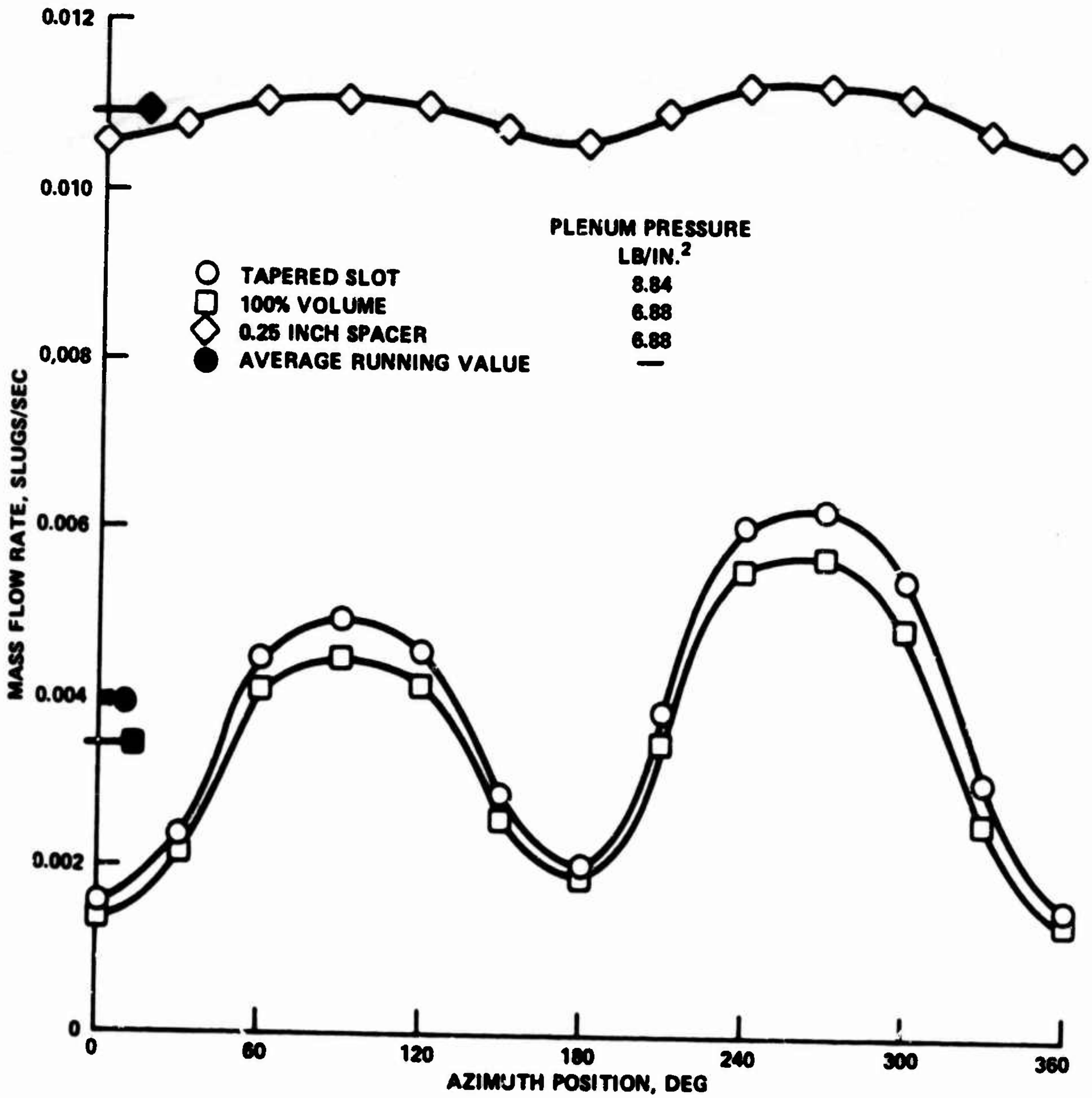


Figure 15b – Two-per-Rev Cam Configuration

Figure 16 – Peak-to-Peak Pressure versus Rotational Frequency for Various Configurations and Plenum Pressures

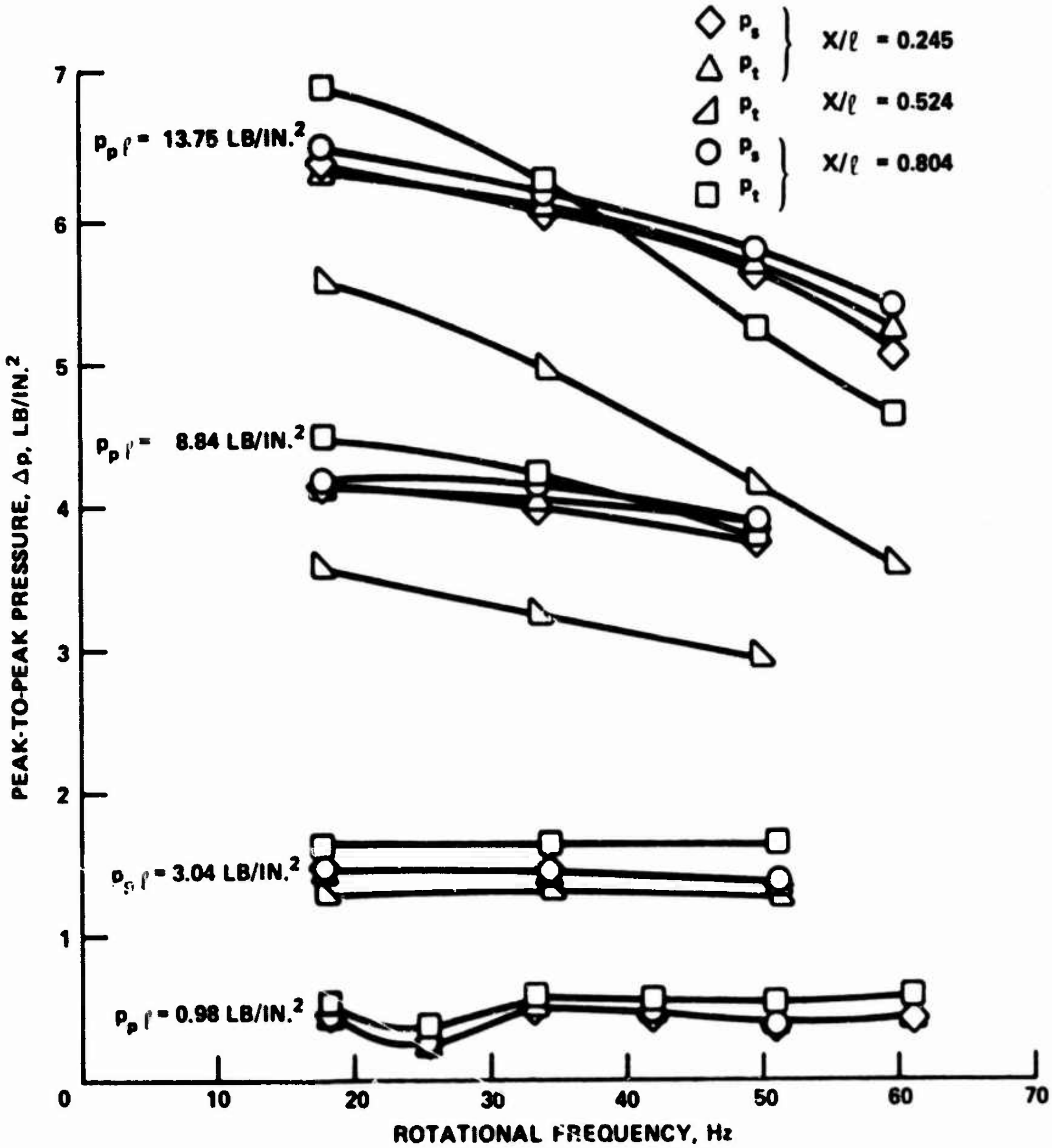


Figure 16a – 100 Percent Volume, Straight Slot, One-per-Rev Cam

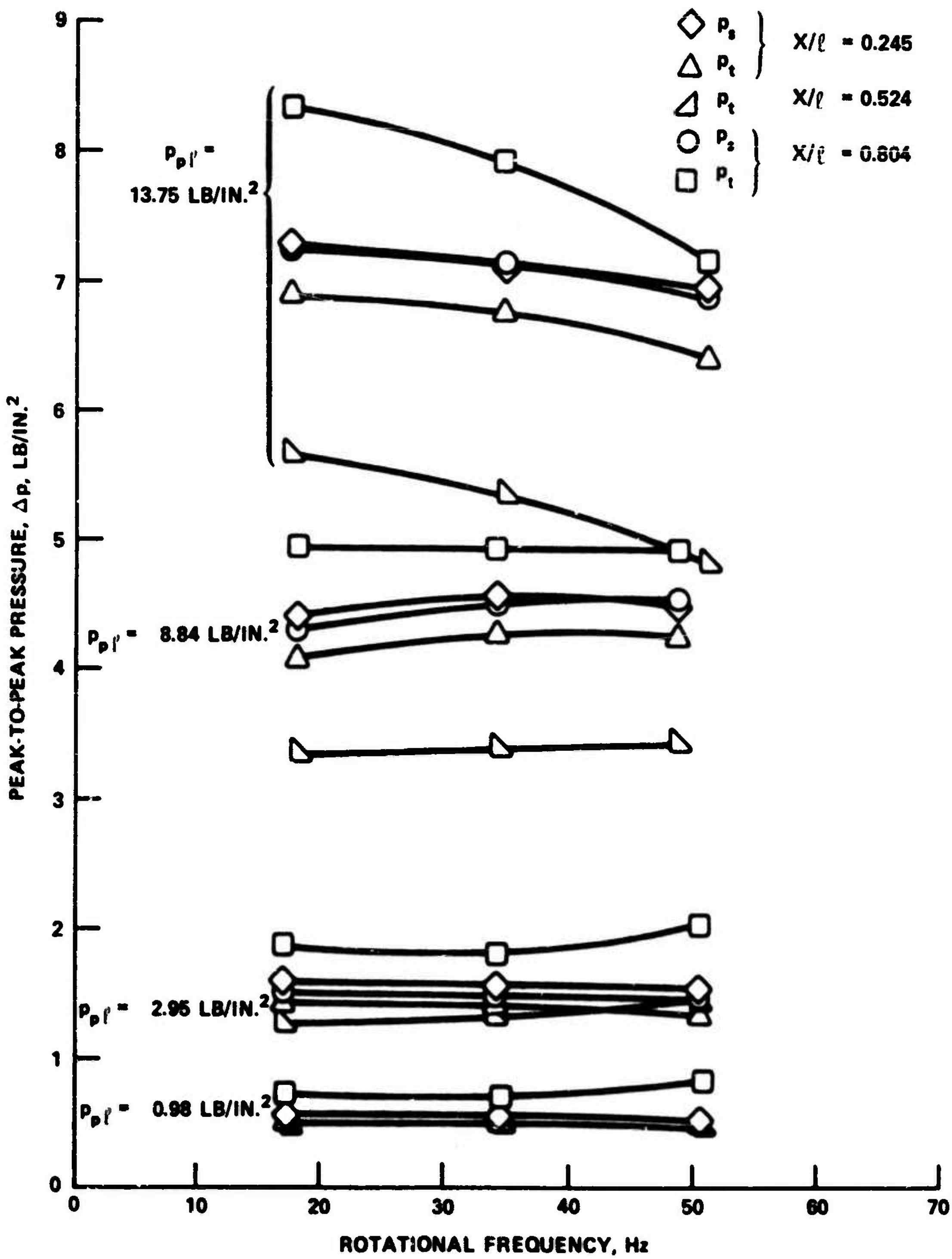


Figure 16b – 67 Percent Volume, Straight Slot, One-per-Rev Cam

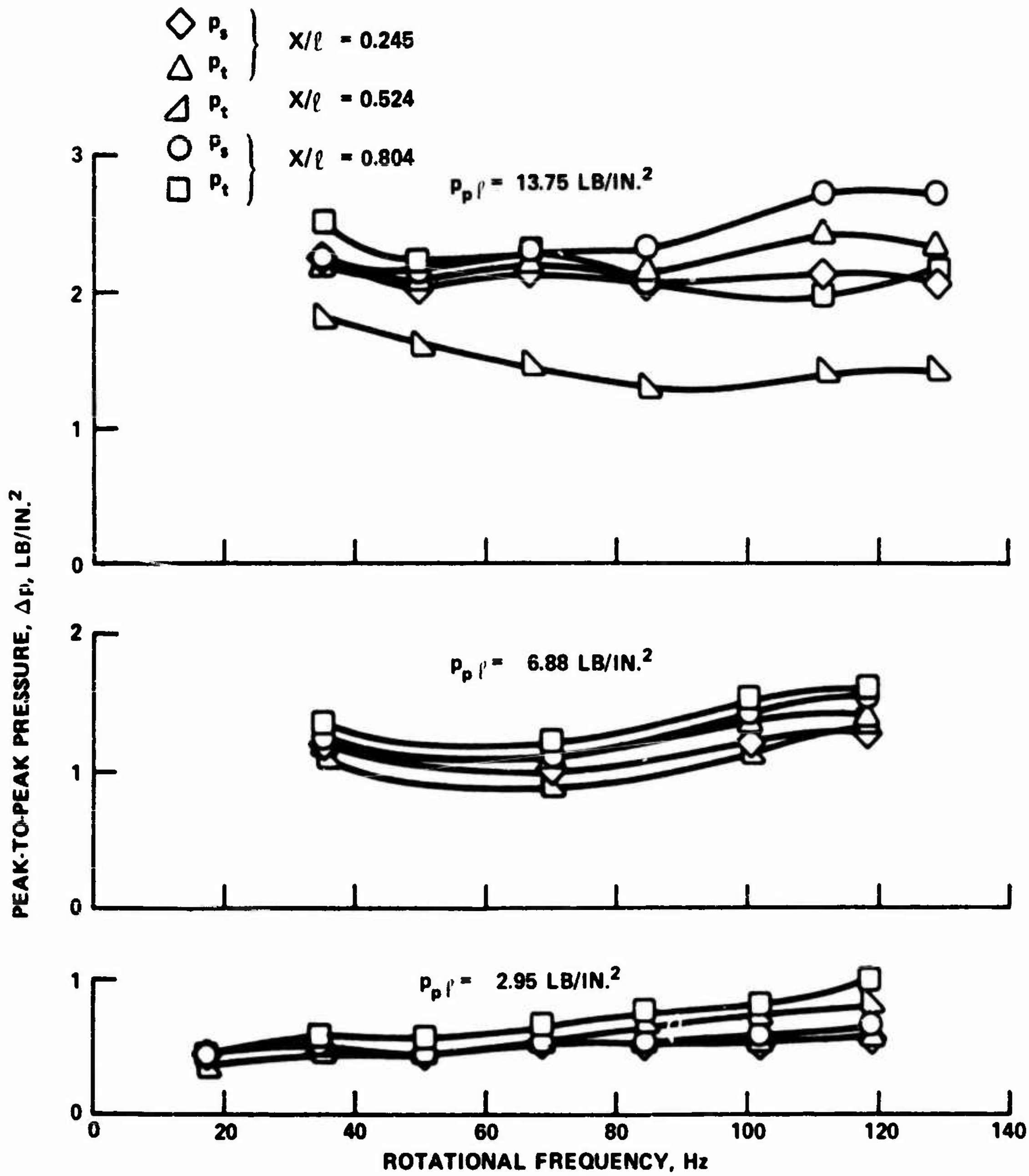


Figure 16c – 100 Percent Volume, Straight Slot, Two-per-Rev Cam

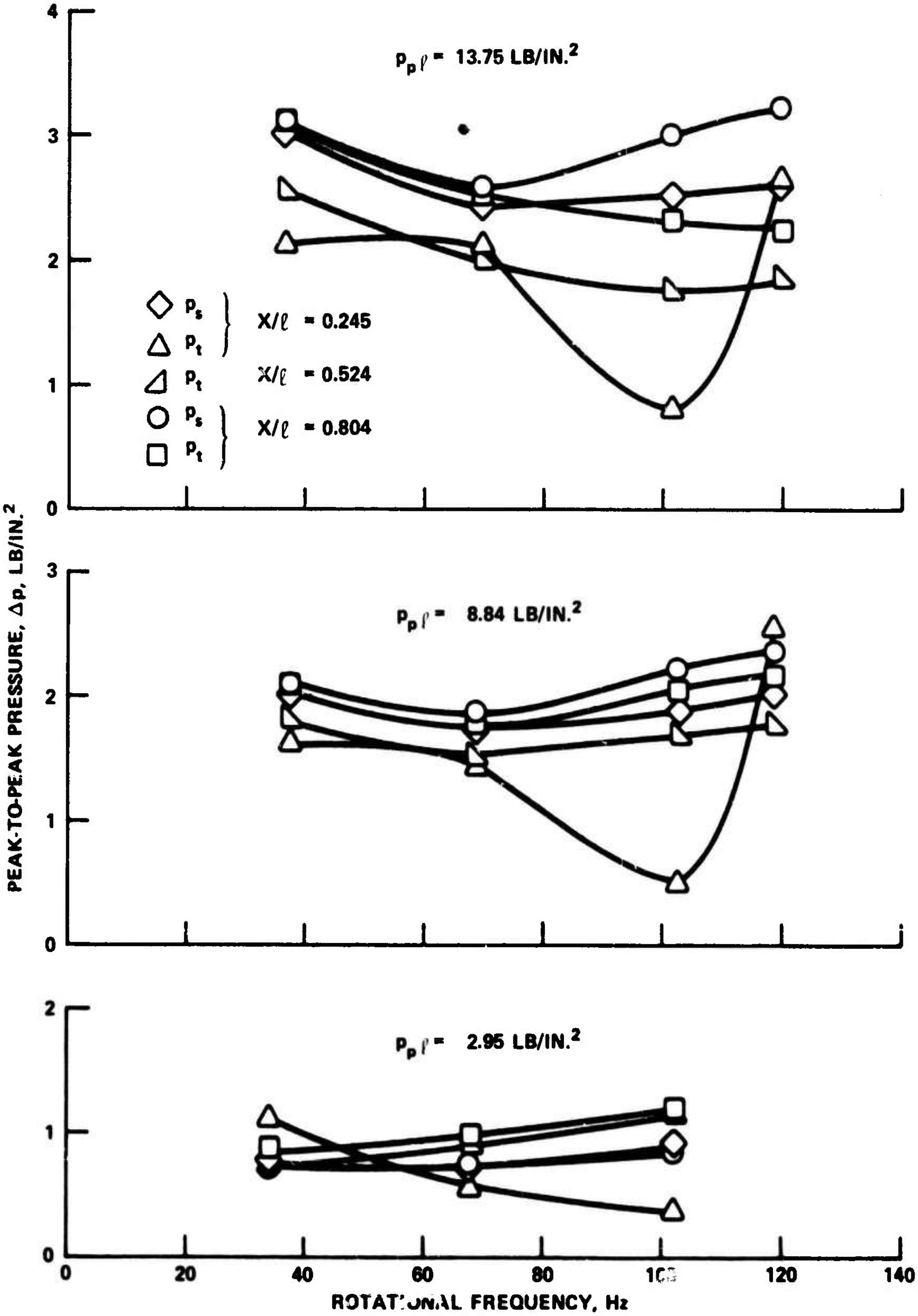


Figure 16d – 100 Percent Volume, Tapered Slot, Two-per-Rev Cam

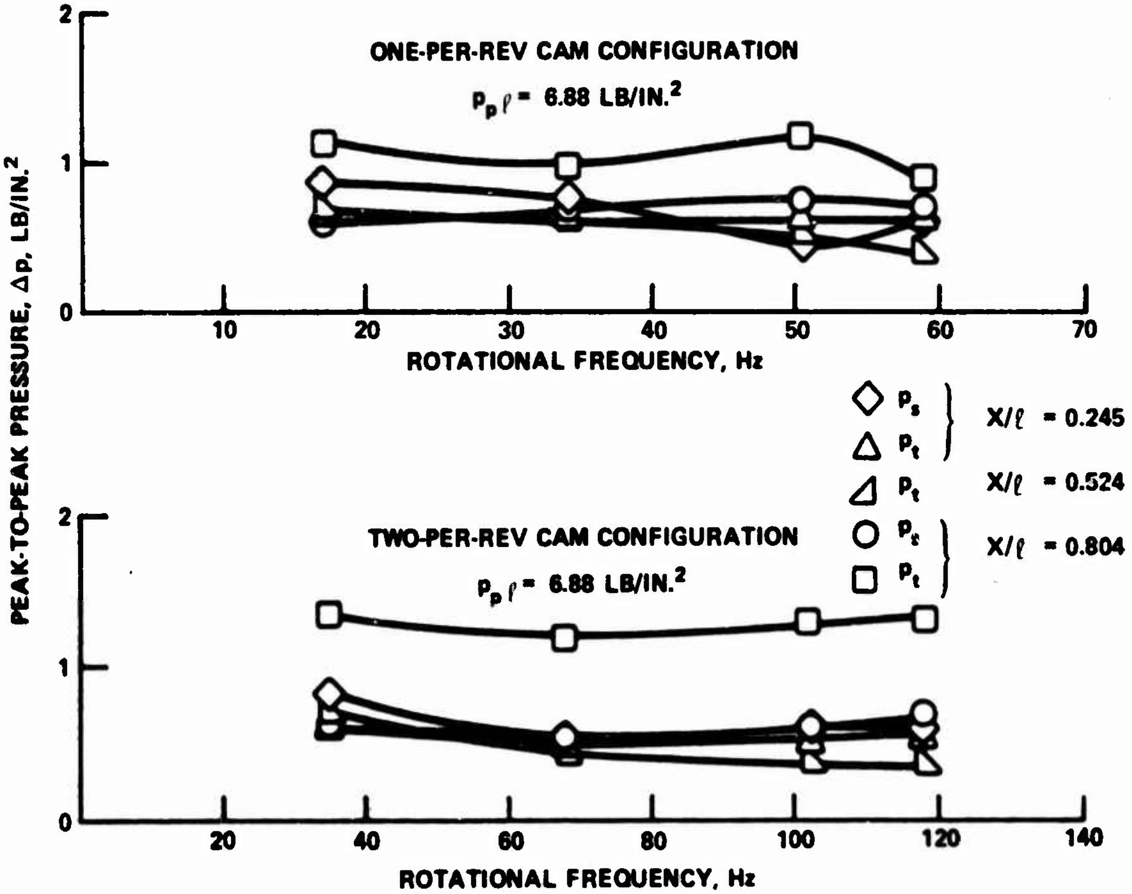


Figure 16e – 100 Percent Volume, Straight Slot, 0.25 Inch Spacer

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1. ORIGINATING ACTIVITY (Corporate author) Naval Ship Research and Development Center Bethesda, Maryland 20034	2a. REPORT SECURITY CLASSIFICATION UNCLASSIFIED 2b. GROUP

3. REPORT TITLE

EVALUATION OF A PNEUMATIC VALVING SYSTEM FOR APPLICATION TO A CIRCULATION CONTROL ROTOR

4. DESCRIPTIVE NOTES (Type of report and inclusive dates)

5. AUTHOR(S) (First name, middle initial, last name)

Kenneth R. Reader

6. REPORT DATE	7a. TOTAL NO. OF PAGES	7b. NO. OF REFS
May 1973	44	3

8a. CONTRACT OR GRANT NO. b. PROJECT NO. F41.421.210 Work Unit 1-1690-100 c. d.	9a. ORIGINATOR'S REPORT NUMBER(S) 4070 9b. OTHER REPORT NO(S) (Any other numbers that may be assigned this report)

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11. SUPPLEMENTARY NOTES	12. SPONSORING MILITARY ACTIVITY
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13. ABSTRACT

A cam-type pneumatic valving system has been developed to provide helicopter rotor control/trim forces. This valving system provides both first and second harmonic rotor control by means of modulating both blade pressures and mass flow rates. Data are presented for (1) constant, one-per-rev, and two-per-rev air modulation, (2) constant and tapered slot distributions, (3) two pipe volumes, and (4) three cam-nozzle gap distances.

The present study demonstrated that air pressure and mass flow rate can be modulated by means of a simple cam valve system. As the gap between the periphery of the cams and the nozzle was increased for a given cam geometry, the mean pressure and mass flow rate increased and the peak-to-peak pressure and mass flow rate decreased. It was also demonstrated that a smooth transfer of the total pressure and mass flow rate occurred in going from a one- to a two-per-rev component (or vice versa)

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14 KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
Circulation Control Rotors						
Circulation Control Rotor Control Trim Forces						
Pneumatic Valving System						
Cam Type Valving System						
Pressure Modulation (constant, one-per-rev and two-per-rev components)						
Mass Flow Rate Modulation (constant, one-per-rev and two-per-rev components)						